STEAM HEATING

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STEAM HEATING

A Manual of Practical Data

Compiled by

THE GENERAL ENGINEERING COMMITTEE

OF

WARREN WEBSTER & COMPANY

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FOREWORD

THE subject of Heating and Ventilation has been covered broadly in many handbooks that are available for reference, but there has been a demand also for a book of information confined exclusively to Steam Heating and covering that field in all necessary detail.

Steam Heating is therefore the one topic of this volume and the editors have aimed to cover the subject with comprehensive data, arranged in such convenient and useful form as will best meet the needs of technical men in the engineering and contracting fields.

The information given is authentic, being based upon actual practice and largely upon the experience of Warren Webster & Company, who, as pioneers, have specialized for more than thirty years in the effective use of steam for all heating purposes. Many of the designs and methods originated by this firm are now the recognized service standards.

Special articles and helpful suggestions have been contributed by John A. Serrell, by the General Engineering Committee, and by John B. Dobson, Ralph T. Coe, William Roebuck, Russell G. Brown, Harry E. Gerrish, Howard H. Fielding, George A. Eagan, E. K. Lanning and other members of the Webster organization.

"Steam Heating" offers the best thought of this organization, and as part of Webster Service, it is intended to be of real value throughout the profession. The observance of good judgment and painstaking care in following its teachings will do much toward obtaining creditable and satisfactory results.

If further explanations, additional information or helpful co-operation are desired, the Engineers and Service Men in the branch offices of Warren Webster & Company throughout the country are always available for consultation and assistance.

CAMDEN, NEW JERSEY
JANUARY 1, 1922

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NOTE.—For convenient reference, each table, illustration and formula is given a compound number, the first part of which indicates the chapter and the second the sequence in that chapter. *Example:* Table 3-6 indicates the sixth table in Chapter 3.

Part I—Steam Heating

CHAPTER I

Elements of Steam Heating

THE purpose of a heating system is to warm the interior of a structure to a desired degree of temperature and to maintain this condition against a lower exterior degree. It is usual to assume the exterior temperature to be the average minimum expected in the locality.

To warm the interior and to maintain a given temperature, heat is required to replace that which is absorbed by the contents and that transmitted through the structure to the exterior.

The unit measure of heat in English-speaking countries is the British thermal unit, which is the heat necessary to raise the temperature of one pound of water from 59 to 60 deg. fahr. This is commonly known as B.t.u., or heat unit.

The quantity of heat required to raise the temperature of a given weight of a substance through one deg. fahr. as compared with the quantity of heat required to raise the same weight of water from 62 deg. to 63 deg. fahr. is called the specific heat of that substance.

The heat content, or quantity of heat per degree of a given mass of a substance, is the product of its specific heat and its weight in pounds.

The rate at which initial heat is required to raise the temperature of a cold structure and its contents to the desired degree in a given time may be much greater than that necessary to maintain the required temperature after initial heating, or warming up, has been accomplished.

The greater the length of time permitted for initial warming, the less difference there will be between the heat requirement per unit of time during initial heating and that required during subsequent maintenance.

Heat losses by transmission through various forms of building structure have been ascertained with more or less accuracy, and much information on this subject has been published from time to time. These data are being constantly improved as new forms of construction appear.

The principal discrepancies between published data on transmission are probably due mainly to various allowances which have been included for infiltration. Infiltration, or air leakage, should be considered independently of structural transmission.

Local differences in workmanship and material of structure, as well as errors in observation, have further contributed to discrepancies, and in many instances the results of tests observed at one temperature difference have been reduced by direct proportion to a "per-degree-difference" basis.

Until recently it has not been generally recognized that this last-mentioned basis is in error, in that it is likely to give too high a rate of heat loss for smaller temperature difference and too low a rate for larger temperature difference than that existing during the test.

The heat transmission factors in Chapter 3 are based upon experience with various substances used in construction under average conditions at a difference of 70 deg. fahr. between interior and exterior temperatures. Factors for other temperature differences are stated as percentages of the 70 deg. normal. The effects of exposure and of varying wind velocities are separately considered as losses due to infiltration.

In order to determine the amount of heat required it is necessary to know or establish:

First: The lowest temperature to which the interior will fall, that is, the "initial" temperature; and the temperature which it is desired shall be maintained within the enclosure, or the "maintained" temperature;

Second: The time period in which it is required that the structure and its contents must be raised from initial to maintained temperature;

Third: The nature and the weight of the building and its contents (especially if large quantities of glass, metal or water are included);

Fourth: The minimum exterior temperature;

Fifth: The direction and anticipated velocities of prevailing cold winds;

Sixth: The construction of the enclosure;

Seventh: The topography of the site, and other local peculiarities.

The heat transmitted hourly through the structure at a temperature difference between maintained interior and minimum exterior temperatures, plus the heat required to warm the infiltrated air through the same difference of temperature, gives the hourly maximum heat requirement during maintenance. In Chapters 3 and 4 these two causes for heat requirements are further discussed.

During initial heating or "warming up," heat units in addition to those required for maintenance must be supplied to raise the temperature of the structure and its contents of air and stored materials from their initial temperature to the desired temperature.

In practice the heat absorbed by the structure and its stored materials is usually neglected, as the error is small. However, if the interior walls or columns are massive, or if the contents of the building include large quantities of materials with high specific heats, such as iron, steel, water, glass, etc., the heat which is absorbed by these must be taken into account.

In almost all cases the heat required to raise the air contents of the enclosure from the initial to the maintained temperature must be considered.

After determining the amount of heat required to warm the various substances during initial heating, the hourly rate at which this additional heat must be supplied during initial heating is obtained by multiplying this heat quantity by the reciprocal of the warming-up period in hours.

Applications of the problem of determining the heat requirements will be found in Chapter 5.

Where the heating requirements for warming-up are large compared with those for maintenance, the radiation necessary for the warming-up requirements and consequently the heat emitted will be correspondingly excessive during maintenance. It is often advisable to increase the length of the warming-up period first allowed in order to reduce this excess radiation.

Overheating after the initial warming-up period, may be avoided by the manipulation of the hand-controlled inlet valves on the radiators or by a system of automatic temperature control.

Having estimated the total hourly heat requirement, the next consideration is the proper proportioning and distribution of radiating surfaces throughout the enclosure, for obtaining the desired heating effect from the circulation of a fluid of higher temperature.

In the following chapters the fluid considered for conveying heat is steam at pressures slightly above that of the atmosphere. The high thermal value, or B.t.u., per pound of steam and the convenience with which it can be utilized by means of commercial boilers, radiating surfaces, pipe and fittings and the special apparatus of the Webster Systems, have demonstrated the superiority of steam at low initial pressures for the great majority of installations.

The radiating surfaces, or radiation, normally used in low-pressure steam heating to transmit heat from steam to the enclosure to be warmed, are of two general classes, Direct and Indirect, each of which has many specific sub-divisions.

Direct radiation, properly classified, comprises only those arrangements of radiating surface which are located directly in the space to be heated.

Radiation which is not wholly exposed in the space to be heated is termed *indirect* radiation. Units which are concealed under window boxes, or in housings having an air inlet near the floor line and a heated air outlet above the radiation, or which are enclosed in casings outside of the space to be heated and which have a cool-air inlet from any source and a warm-air connection to convey by heated air the necessary heat units to the space to be heated, are examples of this type of radiation.

Originally the circulation of air for indirect heating by the method last mentioned was induced entirely by the difference in weight of the air columns before and after coming into contact with the enclosed radiating surface. Present usage designates such surfaces as *gravity indirect*, distinguishing them from surfaces used in the later development, where additional circulating velocity is imparted mechanically by a fan or blower. Where mechanical means are used these surfaces are now designated as *mechanical indirect* or *blast* surfaces.

Certain forms of radiating surfaces exposed in a room and so arranged with dampers and ducts that air wholly from the room or partly from without may be used to convey heat from the surface of the radiator to the room, are called *direct-indirect* surfaces.

The rate of heat transmission through radiating surfaces from a given interior to a given exterior temperature varies not only with all classes of radiation but with all sub-divisions of those classes. This is due mainly to variation in convection, that is, in the facility for absorption of heat from the outer surface into the surrounding medium, and, in a lesser degree, to the dispersion of radiant heat. So great is this variation that, under similar conditions of location and temperature difference, and even in the simplest form of direct radiation, a low, narrow radiator gives off 40 per cent more heat per square foot of radiation than one that is extremely high and wide.

The term "square feet of radiation," therefore, means nothing specific and should not be used indiscriminately for sizing boilers, mains or other apparatus in the heating system.

The radiating surface for the local conditions, heat requirements and architecture, having been selected and located, the proper size of radiating units should be determined. For this purpose the information on heat emission of various types of radiation, Chapter 6, will be found useful.

The pipes which convey the heating fluid from its source to the radiating surfaces are termed supply mains. Those conveying the products of condensation from the radiating surfaces to the point of disposal are termed return mains. The vertical parts of these mains are usually called risers, to distinguish them from horizontal runs. Risers, in turn, are classified by their direction of flow, as *up-feed* or *down-feed* risers. The small branches to individual units of radiation are known as run-outs; those supplying several units as branches, and those conveying all of the heating medium are usually termed trunk mains.

The flow of the heat-carrying medium is always toward a lower pressure, and if the medium is steam confined in pipes or ducts sealed from the atmosphere, the arbitrary dividing line conventionally drawn between pressure and vacuum does not enter. The problem involves only heat content, density, difference in pressure, condensation and friction.

If the lowest terminal pressure in the system is that of the atmosphere as in an open-return line or modulation system, the initial pressure must be somewhat above that of the atmosphere. The amount of pressure above atmospheric depends largely upon the friction which must be overcome in the piping and upon the pressure necessary to give the steam its initial velocity. If, however, a terminal pressure lower than that of the atmosphere is mechanically maintained, as in vacuum systems, the initial pressure may be above, at or below that of the atmosphere as best meets the local conditions.

Vacuum system practice, with few exceptions, demands that a steam pressure at least equal to that of the atmosphere be maintained in the run-outs most distant from the source of steam supply, in order to avoid the in-leakage of air that would otherwise probably occur through minute leaks. This terminal pressure requires an initial pressure higher in some degree than that of the atmosphere. Local conditions, such as source of supply, length and character of pipe run, and use and permanency of the plant, make the selection of pressure difference one of good engineering judgment rather than the application of any fixed rule. The proper basis for proportioning the supply system is dealt with in Chapter 11.

The primary function of return mains is the removal and disposal of the products of condensation. These mains should provide gravity flow wherever possible. Pressure difference should be used to stimulate flow only where gravity alone is not practical.

The products to be removed consist of water, air, vapor, gases from impurities and last, but not to be overlooked, dirt and foreign matter.

The last consists of initial impurities such as core-sand, gravel, chips, mill scale, grease, etc., left in the heating system when erected, together

with rust particles and scale from impure feed water. Were it not for the dirt which collects and the uncertainty as to its volume, return mains might be made much smaller.

Formulæ and tables of capacities of straight, smooth pipes laid on even grades for return of condensation, and tables of accepted capacities compensating for uncertainties of grade and dirt are given in Chapter 11.

The hot distilled water should be returned to the boiler wherever possible. The saving due to the heat content of this water and its freedom from scale-forming and other impurities, warrants considerable initial outlay in return apparatus.

No specific type of return apparatus will best fit all conditions. The single low-pressure heating boiler may have its water line so located that the water of condensation will flow back into the boiler by gravity against the highest steam pressure carried. Between this simple case and a modern high-pressure central generating plant, where part of the exhaust steam is used as a by-product for heating purposes in an extended group of buildings, there is a wide range of conditions. The selection of the best combination of return apparatus for the individual plant is therefore dependent upon comprehensive practical experience.

Some of the possible combinations of return apparatus are described and shown in typical diagrams in Chapter 13, and basic rules are given for estimating proper sized apparatus. However, it is manifest that discussion in this volume cannot cover all requirements, and in this, as in the selection of all apparatus for special conditions, it is recommended that specific engineering advice be obtained from the home office or a nearby branch of the manufacturer, before a selection is made.

Fig. 2-1. Lowest recorded temperatures. United States Weather Bureau statistics.

CHAPTER II

Basic Data Required for Design of a Steam Heating System

INTELLIGENT design of any heating system in either new or existing buildings requires that certain basic data shall be available. For existing buildings the present use of which will continue, it is usually possible to obtain quite definite data to work upon. If plans are the available information, much of the necessary data must be based upon assumptions of probable conditions.

In any event, good judgment, preferably founded upon ripe experience, must play its equal part with scientific knowledge in the final application of the data obtained.

Topography: The design of an efficient heating system, especially where a group of buildings is being considered, requires that a careful study be made of the grade levels of the different buildings, each one to the other, so that, if possible, the condensation from the heating surfaces may flow by gravity to a central point from which it may be returned to the source of steam supply.

In cases where the conditions are such that the condensate will not flow by gravity to a central point, special methods for lifting the condensate to a higher level are necessary as described hereafter.

Location and Character of Source of Heat: It follows from the above that wherever possible the source of steam supply should be located at a lower level than that of the buildings to be heated.

In a plant consisting of a group of buildings there is usually a power generating plant, the by-product from which, in the form of exhaust steam, should be utilized to the fullest extent in the heating of the buildings. The economies incident to the use of this exhaust steam as a by-product frequently determine the adoption of an isolated power generating plant rather than the purchase of power from outside sources and the installation of a boiler plant for heating purposes only.

Exposure and Protective Conditions: By exposure is meant the relation of the outside surfaces of the building or buildings to the prevailing cold winds of winter, which by their pressure cause infiltration of excess quantities of cold air and the rapid removal of heat from the outside surfaces of the structure. To offset this, a larger amount of radiation must be provided on the sides having greatest exposure, than for sides more favorably located with the protection of surrounding buildings or hills.

Consequently the designer should determine the direction of the prevailing winter winds and their probable velocities and duration as well as the topographic features which may afford protection.

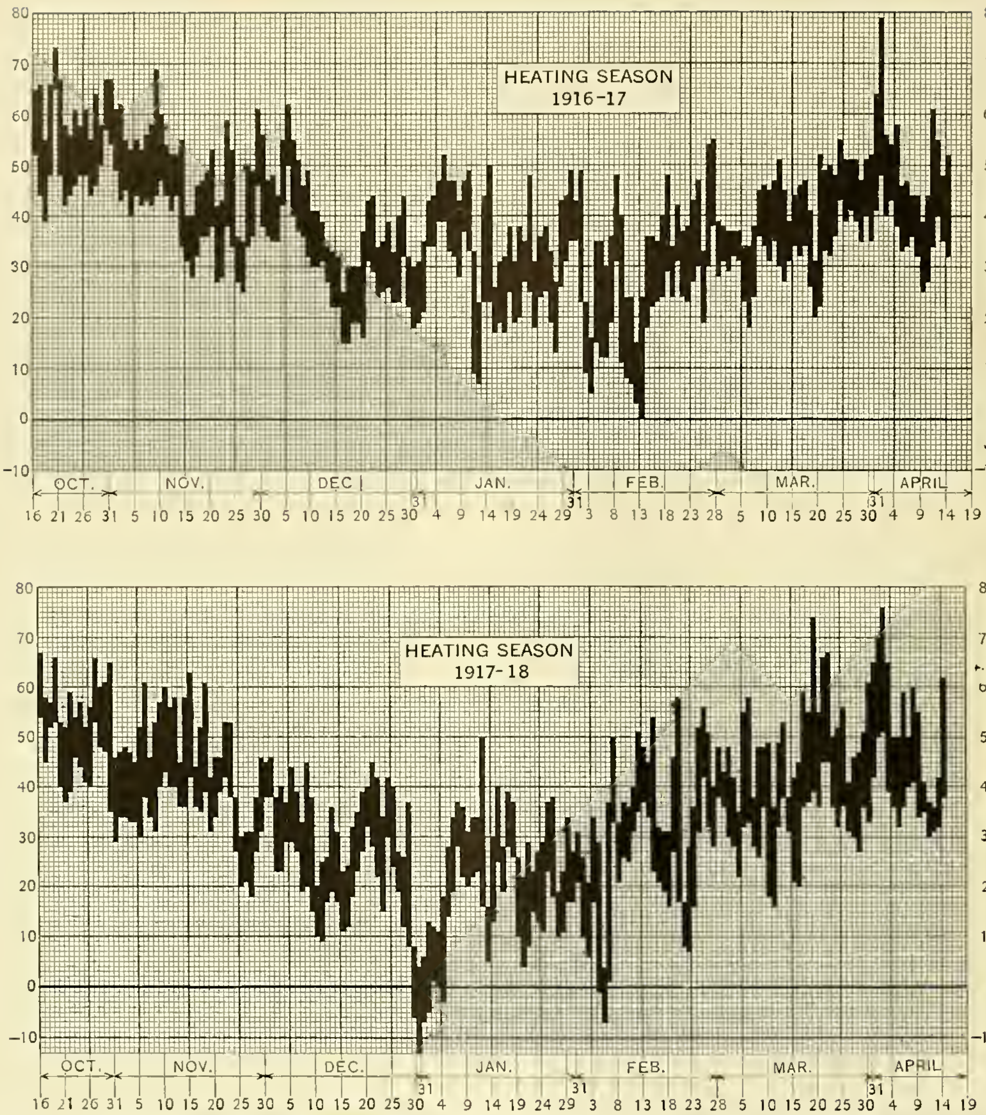
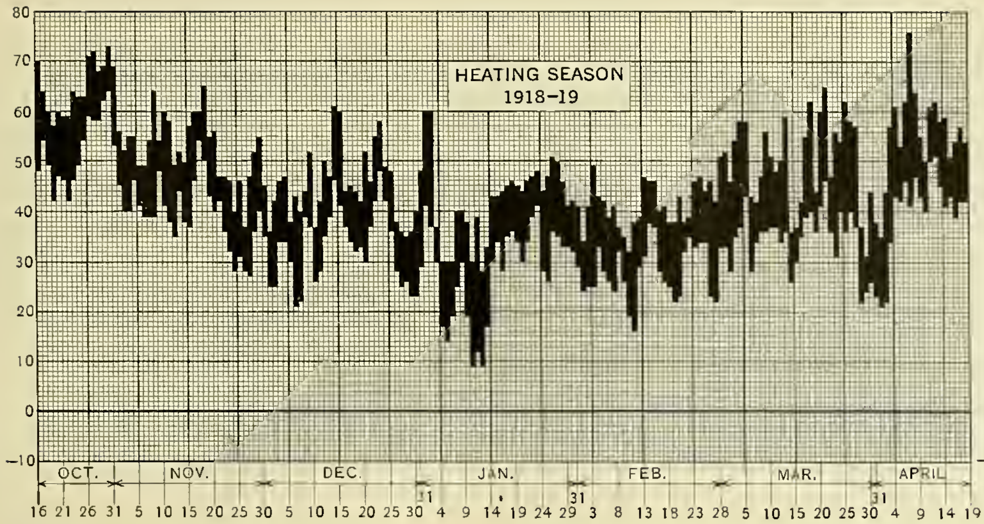
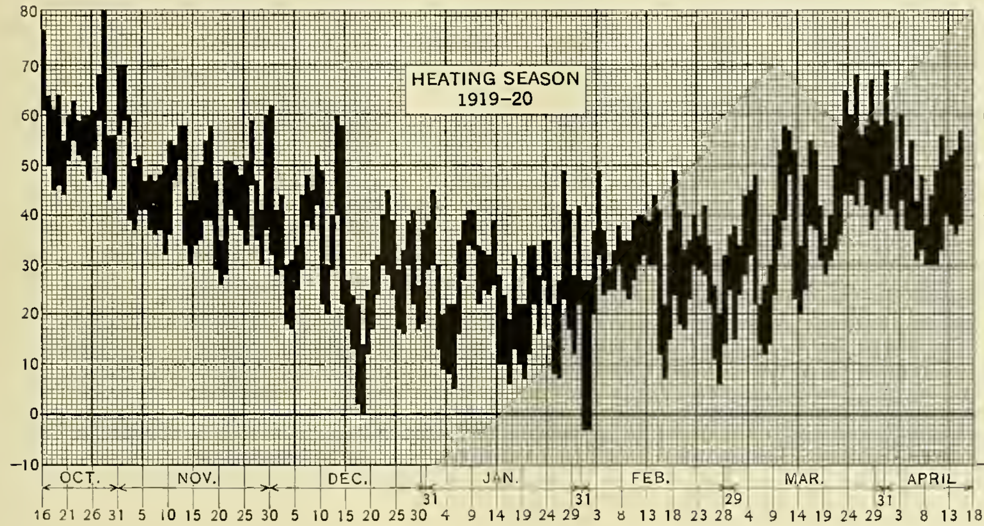


Fig. 2-2. Daily maximum and minimum temperatures in New York City during the heating seasons of 1916-1917 and 1917-1918, (on opposite page) 1918-1919 and 1919-1920. Based upon United States Weather Bureau Reports.

Outside Temperatures: Although the records of the United States Weather Bureau (See Figure 2-1) may show an extreme minimum temperature much lower than that usually experienced in a given locality, it is not customary to estimate heating requirements with that extreme temperature as a basis.

Generally, the average minimum temperature, obtained from United States Weather Bureau records over a period of ten years or longer, is the fundamental consideration. To illustrate the necessity for considering a period of years, rather than to establish the basis on the result of two or three years, charts (Figure 2-2) have been prepared showing the minimum





and maximum temperatures for each day of the heating season for the winter months of 1916-1917 to 1919-1920 for New York City.

These charts show the extreme variation of minimum temperature for different winters and indicate that a safe average cannot be obtained without having records of a long period of years for consideration. They are shown also as a suggested form for the preparation of similar data from Weather Bureau reports for any locality where it is desired to study the temperature conditions upon which the design of a heating system is to be based.

It is possible to operate the most effective types of steam heating systems with a slight increase in steam pressure, which results in an increased rate of heat emission from the radiating surfaces. This flexibility is advantageous during short periods of very cold weather.

Floor Plans, Elevations and Cross-Sections: To properly design the heating system for one or more buildings, complete floor plans and sufficient elevations and cross-sections, showing details of construction, materials, etc., must be available for accurately calculating the heating requirements.

In designing heating systems for existing buildings, accurate data may be obtained by survey, but with designs of new buildings certain assumptions are necessary. These may or may not be justified when construction is complete. A frequent element of error lies in change from original plans without proper consideration for the effect upon the heating system.

These possible discrepancies in construction and deviation in design from original plans make it quite necessary for the designer of the heating system to place himself on record as to the basic factors of his calculation.

Inside Temperature Requirements: The temperature to be maintained and the lowest permissible temperature, are usually governed by the use for which the enclosure is intended.

Inside temperatures are usually determined at the breathing line and not closer than 5 ft. from the most exposed wall.

The important considerations for decision lie in the following questions:

Is the heat to be maintained continuously 24 hours per day or for stated portions of the total 24 hours?

If intermittent heating, how long a time may be allowed to raise the room temperature to the required maintained temperature?

Through how long a period will heat be shut off and how low may the room temperature become during this closed down period?

The following table indicates the usual range in maintained temperatures desired for various classes of occupancy, but it should be kept in mind that temperature is largely a matter of individual preference so that such a table can be considered only as a guide in the final selection.

Table 2-1. Temperature for Various Rooms in Deg. Fahr.

Bath rooms	75 to 85
Churches	60 to 70
Entrance halls to public buildings	50 to 60
Factories	60 to 70
Foundries	50 to 60
Gymnasiums	60 to 65
Homes for aged	80
Hospitals	72 to 75
Lecture halls	60 to 70
Living rooms	68 to 72
Machine shops	60 to 70
Offices	68 to 72
Operating rooms	70 to 90
Paint shops	80 to 90
Prisons, day confinement	60 to 65
Prisons, night confinement	50 to 55
Public buildings	68 to 72
Schools	70
Shops (stores)	50 to 65
Swimming halls	70 to 75
Vestibules for stores and office buildings	70 to 80

The relative humidity of the atmosphere which is likely to exist in any room or building has a bearing upon the desirable inside temperature.

For a living apartment, a normal temperature of 70 deg. fahr. and relative humidity of 50 per cent (about 4 grains of water vapor per cubic foot of content) is considered by most authorities to be a very satisfactory condition of the air. If the temperature is lower than 70 deg., the relative humidity should be higher than 50 per cent or if the temperature is higher, the relative humidity should be lower if the same effect of comfort to the occupant is to result.

It is usual, however, that the relative humidity is found to be much less than 50 per cent in living apartments heated to 70 deg. fahr. and has been observed to be as low as 28 per cent. With very low relative humidity the effect upon the occupant is a feeling of chilliness even though the temperature may be increased to 78 or 80 deg. fahr. This cooling effect is due to the rapid evaporation of moisture from the occupant's skin, which is brought about by the low vapor pressure of the atmosphere. Conversely, where extremely high relative humidity exists, a temperature of 70 deg. fahr. might feel oppressively hot to the occupant.

Contents and Use of Enclosure: A very important consideration for the designer is that of the materials and machinery within the enclosure, and their capacities for absorbing heat. This has an important bearing upon the permissible time limit for warming up.

Large quantities of material or machinery having a high heat content will prolong the time for warming and will have an opposite effect of retarding the loss of temperature when the heat supply is cut off.

For consideration of this factor, the designer should have details of the weight and substance of each of the various items of machinery and materials. With this data and a table of specific heats of substances such as on pages 342-3, the total heat contents or heat-absorbing capacities which influence the warming-up period can be determined.

Likewise, the designer should determine the total heat given off by the operation of the machinery, motors, lights, etc., although this is not of so much importance in buildings where the temperature requirements are those to be maintained during periods when machines, etc., are not in operation.

In schools, theaters, auditoriums, churches, etc., where large numbers of persons may gather, it is necessary to allow for the heat given off by the human bodies if overheating is to be prevented. In such cases, ventilation is usually required to remove the bodily heat with its excessive humidity.

In manufacturing plants, portions of buildings often require unusual quantities of heat to warm the large amounts of air which replace that drawn from the rooms through exhausting fans on grinders, dryers and similar apparatus. This condition requires a careful investigation of the factors involved in the unusual rate of air change.

Character and Location of Heating Surfaces: The selection of the radiation from a choice of direct, indirect, direct-indirect or blast type depends largely upon the use for which the enclosure is intended, the ven-

tilation requirements, the local building laws, school and labor codes, and other general considerations.

Whether pipe coils, cast-iron wall radiation or column cast-iron radiators are to be used for direct heating is usually a question of availability of materials, cost of installation and the esthetic effect required.

The selection of the type and location of the different radiating units may best be determined by a study of the plans and elevations of the building to be heated.

Location of Supply and Return Lines: In installations of the type for hotels, hospitals, office buildings or other public buildings with finished or decorated walls it is customary to conceal the steam and return risers, and their run-outs to radiators, in the wall and floor construction. In factory installations and other less expensive types of construction these lines are exposed and in many instances they are used as prime radiating surfaces.

In cases where the outlets from the risers are taken below the level of entrance to the radiators it is essential that the run-outs shall be so graded that the condensation will flow back by gravity into the risers regardless of the maximum velocity of steam which may flow in the opposite direction. It is therefore of prime importance that the maximum velocity shall be kept well below that at which the condensation will be swept along with the steam. This important feature of design is discussed in further detail in Chapter 12.

A down-feed system of supply is preferable wherever building conditions will permit, since the condensation will then flow in the same direction and will be assisted by the flow of steam as well as by gravity. This permits the use of smaller supply risers and run-outs due to the higher velocities of steam flow which are permissible.

Return run-outs, risers and mains must grade in the direction of flow of condensation to some low point or points from which the condensation will be returned to the source of steam supply or other point of disposal.

CHAPTER III

Heat Transmission

THE same principle of transmission of heat from a higher to a lower temperature that makes steam heating effective, also functions in the transmission of heat through materials of construction to make such heating necessary.

Heat seeks equilibrium, and consequently there is a transfer of heat from a higher to a lower temperature with greater or less rapidity, depending upon the difference in temperature and the character and thickness of the material through which it flows.

For the purpose of estimating the heat losses from enclosures, numerous tests and deductions from practice have been made to determine the rate of heat transmission through the various types and materials of surfaces used for enclosing space. So many variables enter this problem that it is impossible to predict the heat transmission exactly unless all of the peculiarities of any case under consideration have been previously determined.

Tables of heat transmission, therefore, attempt to provide for average conditions of construction of the enclosing substances. Due regard must be given to the facility with which heat is absorbed and removed from the surfaces of the enclosing substances, and to the heat which is transmitted through them, due to the difference between the temperatures existing at their surfaces, which may be termed "heat head."

This heat head has been considered in many formulae as a constant increase per degree of temperature difference. As the result of tests with the same substance under various temperature differences this deduction has been proved to be incorrect. Higher temperature differences cause greater transmission per degree difference than lower temperature differences.

The probable variation in heat transfer under various conditions of heat head is shown in Fig. 3-1. The rate of transfer for any difference between inside and outside temperatures other than 70 deg. is expressed as a percentage of that at 70 deg. difference.

The discussion of Rates of Heat Transmission in this book recognizes the following fundamental conditions:

(1) The maintained inside temperature is that normally existing at the breathing line (5 ft. above the floor) and about 5 ft. from the wall. The breathing line is more often mentioned hereafter as the datum line.

(2) The basic rate of transmission for any substance is the number of B.t.u. which will be transmitted in an hour through each square foot of surface of that substance when the outside temperature is zero and the maintained inside temperature is 70 deg. fahr.

(3) From the above it will be evident that the basic rate is that which is transmitted at the datum line.

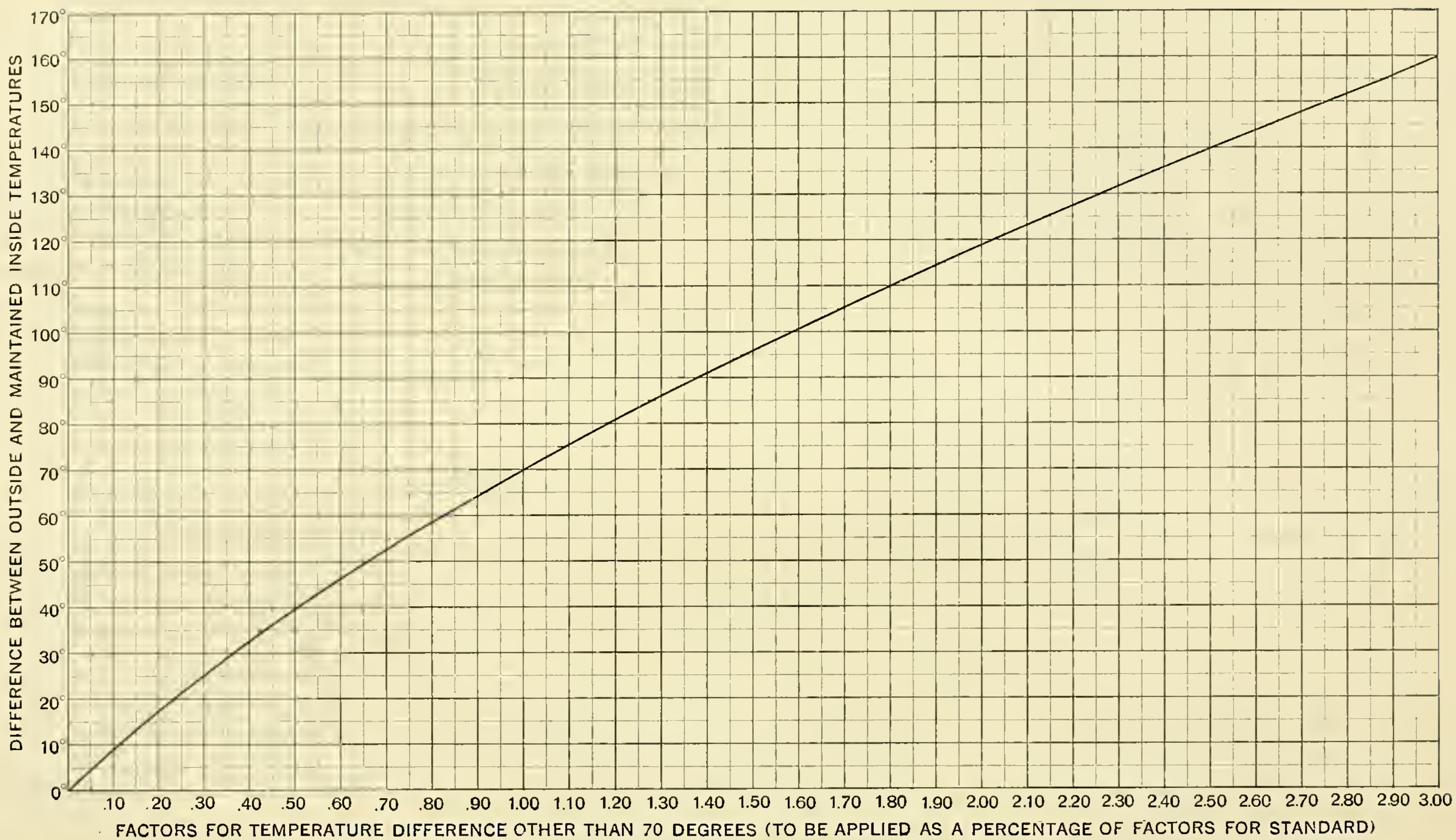


Fig 3-1. Factors for temperature differences other than 70 deg.

In many structures with a ceiling height of 20 to 30 ft., with no mechanical agitation and a low transmission rate through the roof, the average increase in temperature recorded above the datum line to a point close under ceiling has been fully 1 deg. fahr. per ft. In other buildings of similar height with cold roof the average rise has been less than ½ deg. per ft.

The downward circulation set up by the absorption of heat from the air near cold enclosing surfaces tends to agitate the entire contents and reduce the stratification effect. The greater the difference between the exterior and the maintained interior temperature, the greater the agitation and the less the heat rise per unit of height above the datum line.

In estimating heat flow, the average height above the datum line for each class of service should be considered. Due to the increase in temperature above the datum line, the transmission rate for each surface will correspond to that of the temperature of the strata at the average height above datum of such surface rather than that at datum line.

Where the space above the ceiling is heated, the temperature of the strata closest to ceiling will be the highest. In such case it is usual to consider the average temperature to be that midway between the datum line and the upper edge of the vertical enclosing surface and obtain from Figure

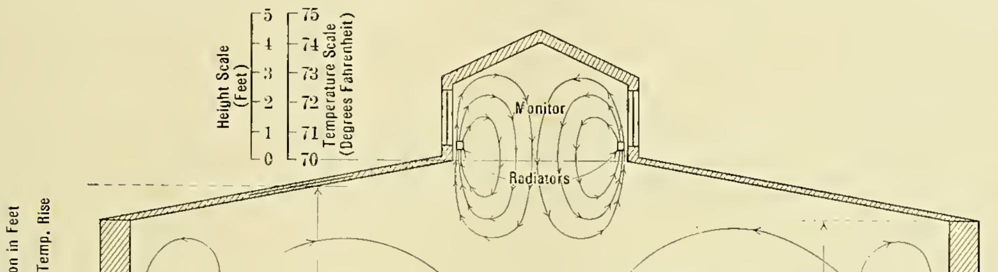


Fig. 3-2. Illustrating heat stratification

3-1, the percentage to be applied to the basic transmission rate of the surface under consideration.

In the case where the space above the roof or ceiling is cold, the temperature of the strata ceases to increase beyond a height somewhat below such roof or ceiling; the distance depending on the rate of transmission through the roof. In this case it is necessary to assume two limits when correcting the basic factor of the enclosing surface to allow for stratification. It is usual to consider the average temperature in this case as that midway between basic level and a level five feet under the cold roof.

The temperature does not always continue to increase in equal amount per unit of elevation above the datum line and in very high rooms the level at which it ceases to increase is likely to be more than 5 ft. below the cold ceiling.

In rooms with a ceiling height over 10 ft. where air is mechanically agitated, there will, in most cases, be a higher average temperature than that at the datum line with a consequent increase in transmission rate; this, however, will be materially less than in cases of similar height where there is no mechanical agitation.

Heat losses through monitors must be specially considered. In such cases it is usual to install heating surfaces within the monitor construction, and for that reason the entire monitor construction should be considered as an individual unit of enclosure with an imaginary floor across the space between the lower edges of its vertical sides.

However, the factors for stratification for figuring heat losses from monitors should disregard the 5-ft. datum line; that is, assuming that the temperature at this imaginary floor line is approximately 70 deg.

In the cases where consideration must be given to the transmission of heat through surfaces at a level beneath the datum line it is advisable to disregard stratification and estimate the heat transmission at the difference between the temperature at the datum line and at the other side of exposed surface.

Basic factors for average height above datum should be fixed on the basic temperature difference of zero outside and 70 deg. fahr. maintained inside.

If the outside temperature for which any particular enclosure is figured is different from zero, or if the temperature to be maintained at the breathing line is more or less than 70 deg., or if both inside and outside temperatures are different from the basic tables, the rates of transmission should be adjusted for the new difference in temperature by factors obtained from Figure 3-1, and applied to all transmission losses through the structure.

It is hoped that in the next edition of "Steam Heating," the result of tests now under way will be sufficiently complete to indicate the probable maximum degree of stratification likely to be encountered in the erecting shops and other structures with high ceilings, which are with increasing frequency presenting their problems to the Heating Engineer.

To obtain the maximum transmission rate due to the average height above the floor of various surfaces mentioned in tables on following pages, the formulae on next page should be employed and a probable maximum value given to S, the rate of heat increase due to stratification.

Windows, doors, walls, and other vertical surfaces

$$T_1 = T + S\left(\frac{H}{2} + D - 5\right) \qquad \textit{Formulae 3–1}$$

Roofs, ceilings, or other horizontal or sloping surfaces

Where upper side is cold $\quad T_1 = T + S\,(H_2 - 10)$

Where upper side is heated $\quad T_1 = T + S\,(H_2 - 5)$

in which

T = temperature at the datum line.

T_1 = average temperature due to stratification at mean height of the surface above datum

S = rate of heat increase above datum, in degrees per foot, due to stratification.

H = height, in feet, of upper level of vertical surface above lower edge.

H_2 = average height in feet above floor of nearly horizontal surface.

D = height in feet above floor of lower level of vertical surface.

Basic Factors: Assuming a value of S in Formulae 3-1, T_1 may be found for any given condition and by referring this T_1 to Figure 3-1, the percentage to be applied to the basic rate may be found. If the temperature conditions are other than basic (zero deg. to maintained 70 deg.) the rates of transmission for heights other than basic should be adjusted to the new temperature difference.

The heat transmission values in the following tables have been proven by experience to be approximately correct. These values may need revision when results are published, of tests contemplated by the Research Bureau of the American Society of Heating and Ventilating Engineers.

Table 3-1. Walls, Clapboard

Construction	Basic factor, 0 to 70 deg.
Clapboard on studs, bare	50
Clapboard on studs, with lath and plaster	35
Clapboard and paper on studs, with lath and plaster	30
Clapboard on studs, with 1-in. sheathing, bare	40
Clapboard on studs, with 1-in. sheathing, papered	35
Clapboard, with 1-in. sheathing on studs, lath and plaster	25
Clapboard and paper, with 1-in. sheathing on studs, lath and plaster	20
Clapboard on studs, with brick fill, bare	28
Clapboard on studs, with brick fill, papered	25
Clapboard on studs, with brick fill, lath and plaster	22
Clapboard and paper on studs, with brick fill, lath and plaster	20
Clapboard and sheathing on studs, sawdust fill, lath and plaster	15
Clapboard, paper and sheating on studs, sawdust fill, lath and plaster	10

Table 3-2. Interior Walls

Construction	Basic factor, 0 to 70 deg.
Plaster, wood lath, studs, wood lath and plaster	24
Plaster, metal lath, studs, metal lath and plaster	28
Studs, wood lath and plaster	42
Studs, metal lath and plaster	48
4-In. hollow tile plastered one side	40
4-In. hollow tile plastered both sides	35
2-In. gypsum block plastered one side	45
2-In. gypsum block plastered both sides	42

Table 3-3. Walls, Stucco on Studs

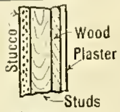
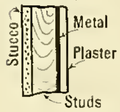
	Construction	Basic factor 0 to 70°
	Stucco on lath, with wood lath and plaster on the inside	40
	Stucco on metal lath, with metal lath and plaster on the inside	45

Table 3-4. Walls, Corrugated Iron

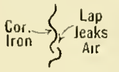
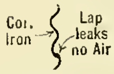
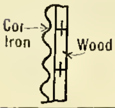
	Construction	Basic factor 0 to 70°
	Plain loose construction on framework	125
	Tight construction on framework	90
	On 1-in. tongue-and-groove sheathing	45

Table 3-5. Walls, Brick

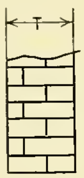
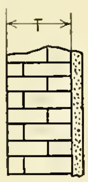
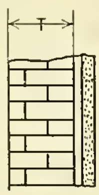
	Thickness in inches T	Basic factor 0 to 70°
	Plain	
	4	45
	8	30
	12	22
	16	18
	20	16
	24	14
	28	12
	32	10
	36	9
	Plastered inside	
	4	40
	8	28
	12	20
	16	15
	20	14
	24	12
	28	11
	32	10
	36	8
	Furred and plastered inside	
	4	30
	8	20
	12	15
	16	12
	20	11
	24	9
	28	8
	32	7
	36	6

Table 3-6. Walls, Hollow Tile Faced with Brick

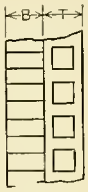
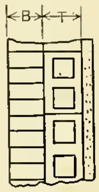
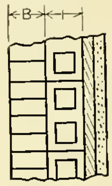
	Thickness in inches Brick	Tile	Basic factor 0 to 70° Plain
	4	4	25
	4	8	20
	4	12	14
	4	16	10
		Plastered inside	
	4	4	20
	4	8	16
	4	12	12
	4	16	9
		Furred and plastered inside	
	4	4	16
	4	8	14
	4	12	12
	4	16	8

Table 3-7. Walls, Concrete Faced with Brick 4-in. Thick

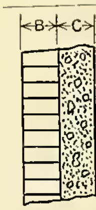
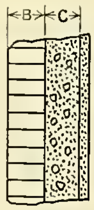
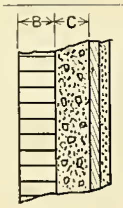
	Thickness in inches Brick	Concrete	Basic factor 0 to 70° Plain
	4	4	35
	4	8	28
	4	12	22
	4	16	18
		Plastered inside	
	4	4	32
	4	8	25
	4	12	19
	4	16	16
		Furred and plastered inside	
	4	4	25
	4	8	20
	4	12	16
	4	16	12

Table 3-8. Walls, Hollow Tile

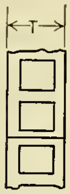
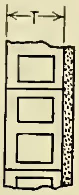
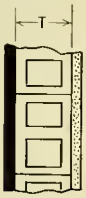
	Thickness in inches T	Basic factor 0 to 70°
	**Plain**	
	4	45
	6	40
	8	28
	10	24
	12	18
	**Plastered inside**	
	4	40
	6	35
	8	25
	10	20
	12	16
	**Stucco, plastered inside**	
	4	35
	6	32
	8	22
	10	18
	12	15
	**Stucco, furred and plastered inside**	
	4	30
	6	28
	8	20
	10	16
	12	14

Table 3-9. Walls, Concrete 4-in. Thick Faced with Stone

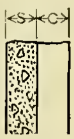
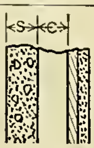
	Thickness in inches		Basic factor 0 to 70°
	Concrete	Stone	Plain
	4	4	50
	4	8	40
	4	12	35
	4	16	27
	**Plastered inside**		
	4	4	45
	4	8	36
	4	12	32
	4	16	24
	**Furred and plastered inside**		
	4	4	33
	4	8	27
	4	12	23
	4	16	18

Table 3-10. Walls, Sandstone or Limestone

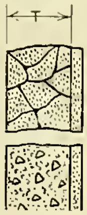
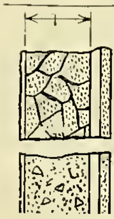
	Thickness in inches T	Basic factor 0 to 70°
	**Plain**	
	4	75
	6	65
	8	55
	10	50
	12	45
	16	38
	20	33
	24	27
	**Plastered inside**	
	4	67
	6	58
	8	49
	10	45
	12	41
	16	34
	20	29
	24	24
	**Stucco, furred and plastered inside**	
	4	50
	6	43
	8	37
	10	33
	12	30
	16	25
	20	22
	24	18

Table 3-11. Walls, Hard Stone or Concrete

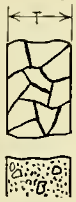
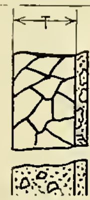
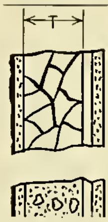
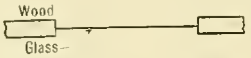
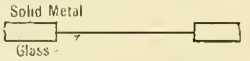
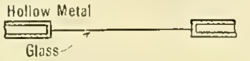
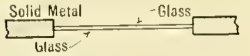
	Thickness in inches T	Basic factor 0 to 70°
	**Plain**	
	4	70
	6	60
	8	50
	10	45
	12	40
	16	35
	20	27
	24	20
	**Plastered inside**	
	4	63
	6	54
	8	45
	10	41
	12	36
	16	32
	20	24
	24	18
	**Stucco, furred and plastered**	
	4	47
	6	40
	8	33
	10	30
	12	27
	16	3
	20	18
	24	13

Table 3-12. Windows

Construction		Basic factor 0 to 70°
	Wood sash, single glazed	75
	Wood sash, double glazed	42
	Solid metal sash, single glazed	90
	Hollow metal sash, single glazed	80
	Solid metal sash, double glazed	65
	Hollow metal sash, double glazed	45

The factors in this table are for transmission rates at the datum line 5 ft. from floor and a temperature of 70 deg. fahr. The temperature T_1 at the centre of a window of any height above the floor will be

$$T_1 = 70 + S\left(\frac{H}{2} + D - 5\right)^\circ$$

Where S = rate of heat increase above datum in degrees per foot, due to stratification.
H = the number of feet of height of the upper edge of window opening above lower edge.
D = the number of feet of height of the lower edge of window opening above the floor.
(See Figure 3–2)

With T_1 established, the factor for correcting the tabular values will be determined from Fig. 3-1. Apply this corrected factor to the entire area of window opening.

Monitors must be considered as separate problems, as if they are structures of themselves with theoretical floors at the level of the base of the monitor. Their transmission losses and the sizing and placing of radiating surfaces should be figured accordingly. The factor should disregard the usual 5-ft. datum line. That is, assume that the temperature at this imaginary floor line is 70 deg. fahr.

Table 3-13. Doors and Wood Partitions

Construction	Basic factor, 0 to 70°
¾-In. to 1-in. thick, tongued-and-grooved	45
1 -In. to 1¼-in. thick, tongued-and-grooved	40
1¼-In. to 1½-in. thick, tongued-and-grooved	35
1½-In. to 2-in. thick, tongued-and-grooved	30
2 -In. to 2½-in. thick, tongued-and-grooved	25
2½-In. to 3-in thick, tongued-and-grooved	20

Table 3-14. Roof Construction

Construction	Basic factor, 0 to 70°
Flat tile on strips	75
Flat tile on sheathing	45
Slate on strips	78
Slate on sheathing and paper	35
Corrugated iron on strips	125
Corrugated iron on sheathing	45
Tin on strips	110
Tin on sheathing	40
Tin on sheathing with paper	30
Shingles on strips	60
Shingles on sheathing	30
Shingles on strips over tar paper and sheathing	15
Reinforced concrete composition 2-in., paper, tar and gravel	50
Reinforced concrete composition 3-in., paper, tar and gravel	45
Reinforced concrete composition 4-in., paper, tar and gravel	40
Hollow tile 4-in., paper, tar and gravel	20
Hollow tile 6-in., paper, tar and gravel	18
Metropolitan 3-in., paper, tar and gravel	20
Metropolitan 4-in., paper, tar and gravel	15
1-In. wood with 5 to 8-ply paper, tar and gravel	20
1-In. wood with felt roofing	25
1½-In. wood with 5 to 8-ply paper, tar and gravel	18
2-In. wood with 5 to 8-ply paper, tar and gravel	15
2½-In. wood with 5 to 8-ply paper, tar and gravel	12
2-In. Federal cement tile, paper and tar and gravel	50

Table 3-15. Roof Glass and Skylights

The surface to be considered is the total surface of glass and frame

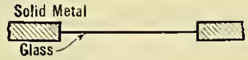
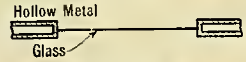
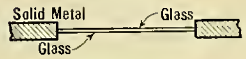
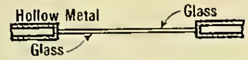
Construction		Basic factor, 0 to 70°
	Wood sash, single glazed	75
	Wood sash, double glazed	42
	Solid metal sash, single glazed	90
	Hollow metal sash, single glazed	80
	Solid metal sash, double glazed	65
	Hollow metal sash, double glazed	45

Table 3-16. Floors Above Cold Space

The factors are for 0 to 70 deg. difference in temperatures. For any other difference, the basic factor should be corrected in accordance with chart, Fig. 3-1

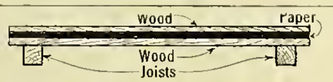
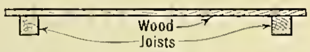
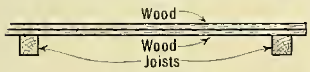
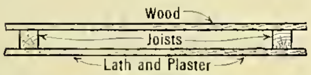
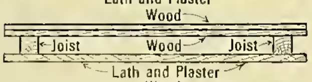
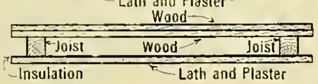
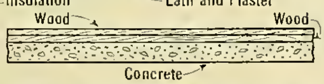
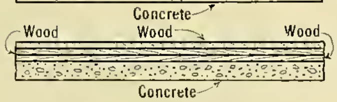
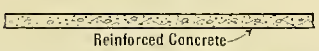
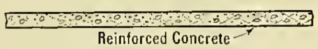
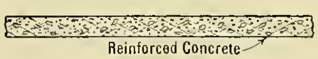
Above cold space	Description	Basic factor, 0 to 70°
	Mill construction, 3-in. wood and paper plus ⅞-in. surface	12
	1-In. single wood floor on joists	25
	2-In. double wood floor on joists	15
	1-In. single wood floor on joists with lath and plaster	14
	2-In. double wood floor on joists with lath and plaster	10
	2-In. double wood floor on joists with insulation and lath and plaster	4
	2-In. double wood floor on 4-in fireproof concrete	6
	1-In. wood flooring on double wood and 4-in. fireproof concrete	4
	4-In. concrete slab, metal reinforced	.70
	6-In. concrete slab, metal reinforced	60
	8-In. concrete slab, metal reinforced	50
	10-In. concrete slab, metal reinforced	40

Table 3-17. Floors Laid on Ground

These factors are for 0 to 70 deg. difference in temperature. It is usual, however, to assume the temperature of the ground beneath the floor as 50 deg. fahr. For this difference the above basic factor must be corrected by means of the chart in Fig. 3–1

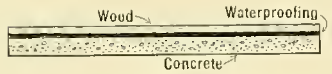
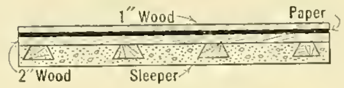
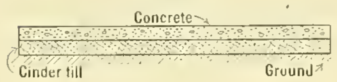
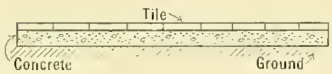
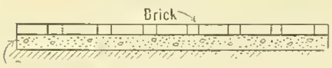
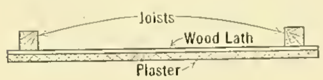
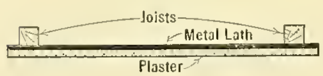
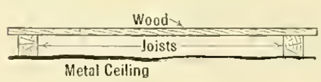
	Construction	Basic factor 0 to 70°
	1-In. single wood floor on wood sleepers	9
	2-In. wood floor on 4-in. water-proofed concrete	7
	3-In. double wood floor with paper between on sleepers in 4-in. concrete	4
	4-In. concrete floor on ground	22
	4-In. concrete floor on cinder fill	20
	1-In. tile floor on 4-in. concrete	22
	2½-In. brick floor on 4-in. concrete	20

Table 3-18. Ceilings

The factors are for 0 to 70 deg. difference in temperatures. For any other difference, the basic factor should be corrected in accordance with chart, Fig. 3-1

	Construction	Basic factor 0 to 70°
	Wood lath and plaster on joists	42
	Metal lath and plaster on joists	46
	1-In. single wood floor on joists with wood lath and plaster	18
	2-In. double floor on joists with wood lath and plaster	14
	1-In. single wood floor on joists with stamped metal ceiling	25

CHAPTER IV

Air Infiltration

WIND blowing against walls causes a leakage of air into the enclosure and an outward leakage from the enclosure through the opposite sides. Additional leakage is caused by temperature difference within and without regardless of wind velocity. These leakages are sometimes referred to as air change, but in this book are called *air infiltration.*

As the air enters and leaves the enclosure at different temperatures, sufficient B.t.u. or heat units must be provided to heat this air between the two temperatures. Air infiltration therefore becomes one of the important factors in the determination of the heat requirements of a room or an enclosure.

Some rules for heat requirements of an enclosure regard that portion due to air infiltration as an additional quantity to be based upon an arbitrary hourly air change or upon a certain percentage of the best transmission factor.

Examination of the air infiltration shows that most of the air leaks are around the doors, windows and other similar openings. The quantity that expresses the heat requirements due to this infiltration of cold air should therefore be based upon the sum of the openings through which this leakage occurs, rather than upon the area of the doors, windows and similar openings of the structure.

Any determination of the quantity of air infiltrated must take into consideration the velocity and direction of the wind in relation to the openings of the enclosure. Where an enclosure has openings on more than one side, the infiltration for all openings must be determined and the radiation for this loss proportioned and located according to the maximum degree of infiltration that may occur on any side. *This method will give an excess of radiation on the sides where leakage is outward, but there is no alternate without having some sides of the room feel cool at some wind direction.*

In small rooms having window exposures on more than one side, and which ordinarily can be heated with one radiator, it is only necessary to consider the infiltration for the side of maximum exposure and locate the radiation on that side.

The leakage in narrow monitors and rooms where cold drafts will not be objectionable may be considered only on the side where maximum wind velocities occur. A portion of the heat to care for this infiltration can then be applied to the other side. Where the wind strikes the surface at an angle, the resultant velocity at right angles to the surface must be considered. This is equal to the actual velocity times the sine of the angle of incidence.

Normally the same maximum wind velocity should be considered on the north and west sides, while on the south and east sides one-half of these velocities may be used except where special wind conditions exist.

A suggested extreme condition for New York and vicinity would be 15 miles per hr. wind velocity with a temperature of zero. Generally low wind velocities prevail at extremely low temperatures.

The many variables make reference to experiment more reliable than attempts to determine theoretically the perimeter air infiltration of windows, doors and similar openings. Little dependable experimental data is available at present, but such as is now obtainable must be used as a basis until better is to be had.

Experiments on air infiltration of windows have been made by using a fan to direct wind velocities against a test window set in the side of a tight enclosure and having an opening for pitot tube readings on the opposite side. Further details regarding some of these experiments by Whitten will be found in the 1908 Transactions of the American Society of Heating and Ventilating Engineers, and others by Voorhees and Meyer in the 1916 Transactions.

Similar tests are being conducted by the Research Bureau of the American Society of Heating and Ventilating Engineers and the United States Bureau of Mines. In these tests natural air velocities are used and the infiltration determined by reduction in carbon-dioxide content of air in the room. A preliminary report of these tests describing the method in detail was given by Mr. O. W. Armspach in the Journal of American Society Heating and Ventilating Engineers, January, 1921.

Figure 4-1 gives the approximate leakage in cubic feet per minute per lineal foot of sash perimeter for double-hung locked windows, with and without metal weather strips.

The type and construction of the windows to be used should be definitely

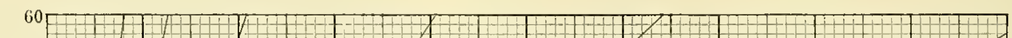


Fig. 4-1. Air infiltration for double-hung windows

known before the infiltration is estimated and this data recorded in a similar manner to data regarding the type of wall, roof or other construction of the enclosure.

Due allowance should be made for special sash. The meeting rail must be considered in measuring the perimeter of double-hung sash.

In windows with steel-section frames properly bedded, only the perimeter of that portion which opens, or the ventilating sash, need be considered.

In industrial plants where it is intended to install mechanical exhaust systems for removing dust or fume-laden air, special means must be provided to care for the corresponding increase in infiltration as described on page 65 of Chapter 7.

For well-fitting doors the average window values can be used, but for sliding and similar poorly fitting doors, as used in industrial buildings, the values for a poor window should be used.

The leakage values as read from Figure 4-1 when multiplied by 60 x 0.0864 (density of air at zero) x 0.2375 (sp. ht. of air), will give the heat units per hour required to warm the infiltrated air from 1 ft. of perimeter, 1 deg. fahr.

Example: Assume an average double-hung window 3 ft. wide by 6 ft. high with a perimeter of 21 ft., outside temperature zero, inside temperature 70 deg. fahr. with wind velocity of 15 miles per hr. Referring to Figure 4-1, the leakage per foot of perimeter is found to be 1.60. Then 21 x 1.60 x 60 x 0.0864 x 0.2375 x 70 = 2893 B.t.u. per hr. required to heat the air infiltration from this window.

The following tables will be found useful in determining the heat units required to care for the infiltration. These values are for plain double-hung windows. If equipped with a good metal weather strip, use 20 per cent of the tabulated values.

Table 4-1. B.t.u. per Hour Required per Lineal Foot of Perimeter for Windows

	Wind vel. Miles per hr.	Infiltration Cu. ft. per min. per ft. of perimeter	Temperature difference inside and outside of enclosure 50°	60°	65°	70°	80°
Good window	5.	.36	22	27	29	31	35
	7.5	.54	32	39	42	45	52
	10.	.72	44	53	58	62	71
	15.	1.08	66	80	86	93	106
	20.	1.42	87	105	114	122	140
Average window	5.	.56	34	41	45	48	55
	7.5	.85	52	63	68	73	84
	10.	1.12	69	83	90	97	110
	15.	1.68	103	124	134	145	165
	20.	2.22	137	164	178	191	219
Poor window	5.	1.07	66	79	86	92	105
	7.5	1.60	95	115	125	134	154
	10.	2.12	131	157	170	183	209
	15.	3.12	192	230	250	269	307
	20.	4.07	251	301	326	351	401

CHAPTER V

Method of Calculating Heat Requirements

CHAPTERS 1 and 2 give the general data that must be known in calculating the heat requirements of any structure. Several rules and formulae have been devised to determine the amount of heat that must be supplied to maintain a room or enclosure at a predetermined temperature with a known surrounding temperature. Many of these formulae were derived when construction, size of window opening, etc., were similar and are not flexible enough to cover the problems of today.

If the air within an enclosure is maintained at a temperature higher than that surrounding, there must be a natural transfer of heat through the enclosing structure to the air of lower temperatures. This transfer may be to the air outside, to any adjoining rooms and to air above and below, if these are at lower temperature than that in the room.

To heat the enclosure to and maintain it at a predetermined temperature, an equal amount of heat must be supplied at the rate at which it is transferred. The most accurate method of determining the quantity transferred is to determine the hourly rate of heat transfer from the heated enclosure to the surrounding air. This quantity is usually calculated in British thermal units per hour; that is, on the B.t.u. basis.

The total quantity transferred is made up of four principal heat requirements.

The first is the heat required to warm to the desired inside temperature, the air that leaks in through the various openings around the window and door perimeters, etc., from the outside. To calculate the heat units for these requirements, the width and lineal feet of the openings, and the wind velocity against the side of the enclosure where the openings are located, must be found, and with these data the air infiltration determined. The product of the air infiltrated in cubic feet per hour, the density of the air, its specific heat and the difference between the inside and outside temperatures will give the heat required per hour for infiltration. This subject is further discussed in Chapter 4.

The second is the heat transmitted through the various materials of which the enclosure is constructed. To calculate this requirement, the area, thickness and kind of the various materials through which this transfer occurs, and the temperature difference between the air on the two sides of the material must be known.

The product of the area of any material in square feet, the transmission coefficient for that material in B.t.u. per hour, and the difference between the inside and outside temperatures will give the heat transmitted per hour through that material. The sum of quantities so found for all materials of the structure is the total loss of heat from the enclosure by transmission.

A desired maintained interior temperature of 70 deg. fahr. and a minimum external temperature of zero have been adopted in this book as a

standard. All transmission coefficients, therefore, are given in B.t.u. per hour per square foot of surface for this temperature difference, with correction factors for other differences.

A table of these factors for various materials used in building construction will be found on pages 25 to 30.

A third requirement enters into the calculation where the heating is not continuous. This may be referred to as a heating requirement, or the heat necessary, to raise the air of the enclosure from its initial temperature to the desired maintained temperature. It is evident that if only sufficient heat is supplied to compensate for the air infiltration and transmission requirements, the temperature of the enclosure would approach but not reach the predetermined temperature, unless additional heat units are supplied for heating an amount of air equivalent to the cubic contents of the space to be heated. To calculate this requirement, cubic contents of the enclosure, initial and final temperatures of the internal air, and time desired to raise the air through this temperature range must be determined.

The product of the quantity of air in cubic feet, the density of the air, its specific heat, and the temperature difference, is the quantity of heat required for initial heating of the air. If this quantity be then multiplied by the reciprocal of the heating-up period in hours, the product will be the quantity of heat that must be supplied per hour during the heating-up period, to supply the heat absorbed in heating the air.

A fourth heat requirement should be included in calculations where the heating is not continuous, and where large quantities of materials such as metal, water, glass, etc., are stored in the enclosure and must be heated like the air contents, from initial to maintained inside temperature.

The product of the weight of such material in pounds, its specific heat and the desired temperature range is the heat absorbed by the material.

This quantity must also be multiplied by the reciprocal of the heating-up period in hours to obtain the hourly heat requirement during initial heating to compensate for this absorption of heat. The longer the heating-up period selected the less will be the difference in the hourly requirements during initial and maintained heating.

Where large quantities of such stored materials are taken into and removed from the enclosure at intervals the heat absorbed by these materials must be considered.

The sum of these four requirements gives the total hourly rate at which heat must be supplied to maintain the enclosure at a predetermined temperature, or to raise the temperature of the enclosure from its initial to predetermined temperature, as the case may be.

Applying this method of calculating heat loss requirements, Figure 5-1 represents the main floor of a residence with warm basement and second floor. Under these conditions, no ceiling or floor loss need be considered.

The quantities taken from the plan and the basic data are entered on the Heat-requirement Computation Sheet, Table 5-1.

The requirements are figured for each exposed side as in Room 1. The requirement for the north side is 12230 B.t.u., for the east side 4830 B.t.u., for the west side 1635 B.t.u. and the B.t.u. required for initial heating

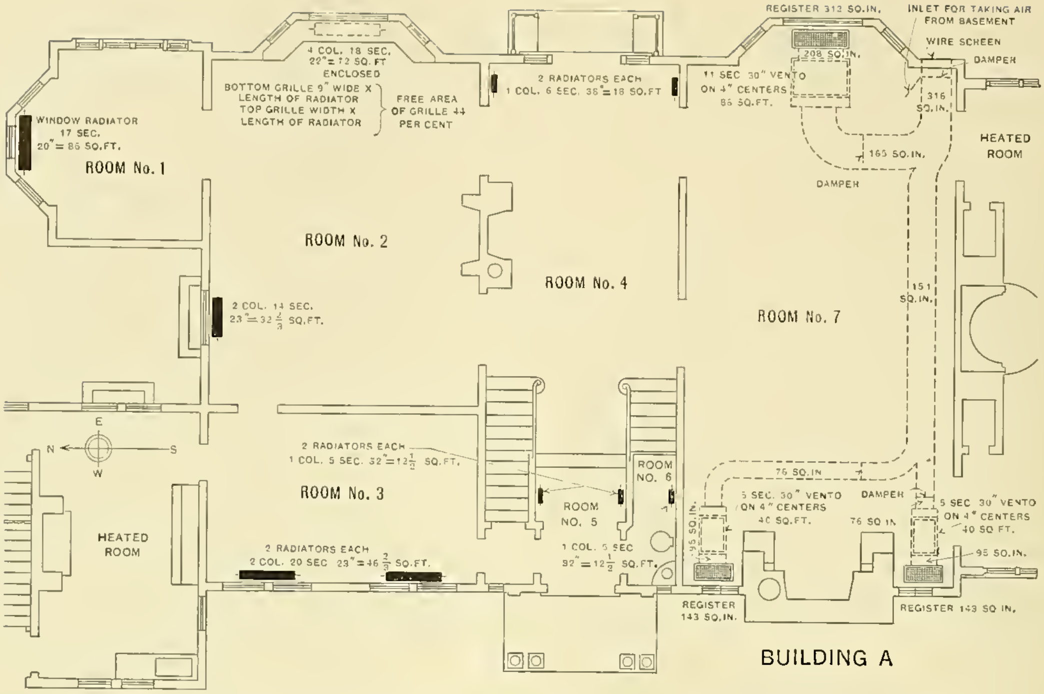


Fig. 5-1. Plan of residence floor used as basis for heat-requirement computation sheet, Table 5-1

of the air contents is 632, making a total maximum requirement of 19,327 B.t.u. per hour. The heat supply for this room should be placed under the north window.

The requirements in B.t.u. per hour as taken from the computation and divided in a similar manner are marked on the plan for each room.

Another illustration of the method of calculation is given in the Heat-requirement Computation Sheets, Table 5-2, for the factory building shown in Figure 5-2.

The calculation has been separately made for the sections as marked in the figure, so that the losses may be proportioned to the exposures.

It will be noted that the correction factors are used to change the 70-deg. temperature-difference coefficients to correct values for the given temperature differences which may vary due to stratification.

In section C, the calculations for the north and south walls with their windows and doors from the floor to line a—b were made separately from the balance of the losses for this section.

As the air infiltration from the upper sash would not be felt directly by the operators in the building, the infiltration has been calculated for only the west or side of maximum wind velocity.

The infiltration factor for the doors has been taken as that of a *poor* window, and in calculating the window infiltration losses only the perimeter of the ventilating portion of the window has been considered.

The requirements for the various walls and sections as taken from the calculations are marked on the drawing in their relative locations.

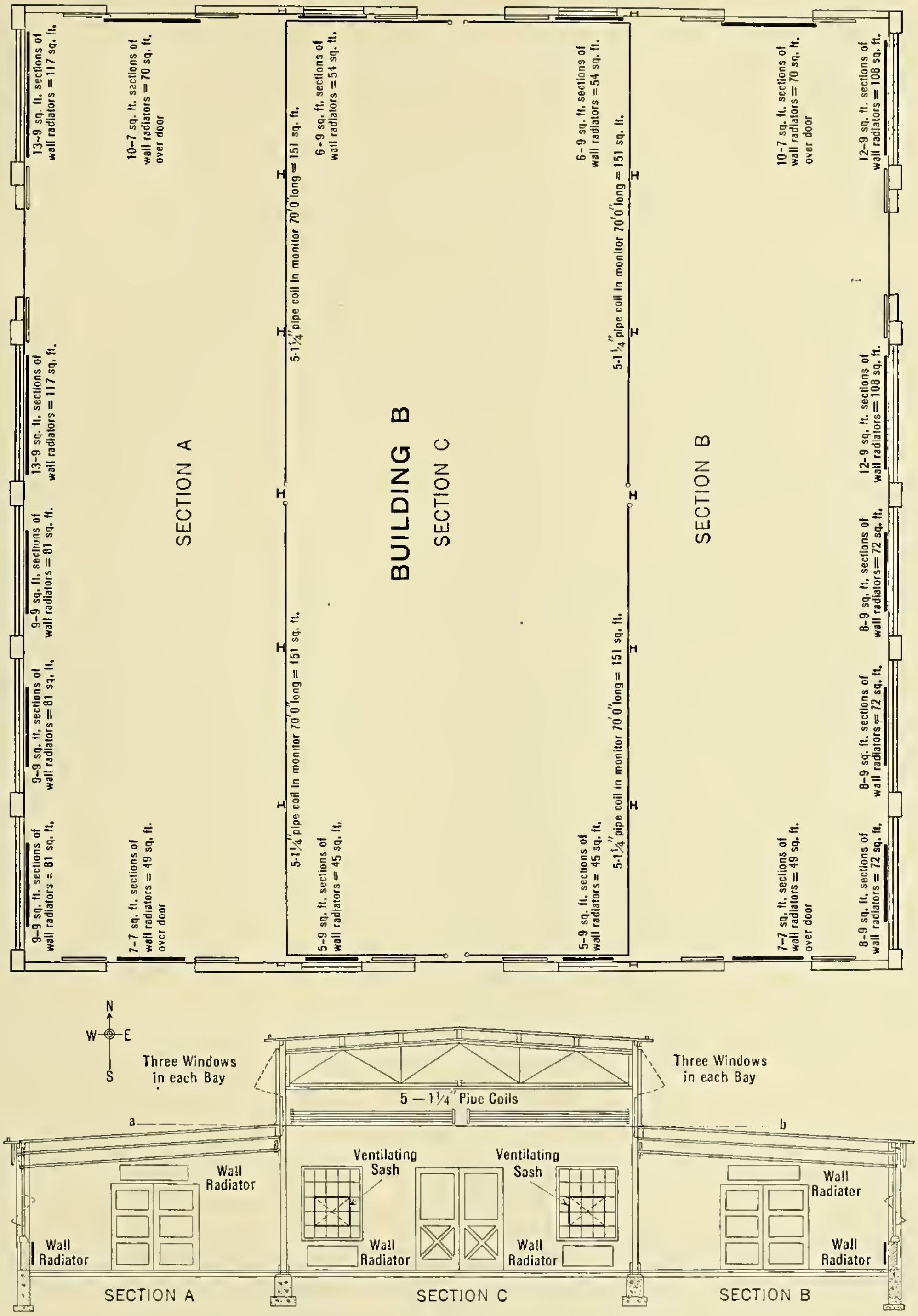


Fig. 5-2. Factory building used as basis for heat-requirement computation sheet. Table 5-2

Table 5-1. Heat-Requirement Computation Sheet

Job name, Building A .... No.	Outside temp. 0°	Wind vel. Miles per hr. N. 15.	Type of Windows Frame double hung, average
Location	Inside " 70°	" " " " " E. 7.5	
Owner	Initial " 50°	" " " " " S. 7.5	Type of doors $1\frac{1}{2}$-in. Wood swing
Architect	Heating-up period Hrs-1	" " " " " W. 15	
Engineer			

Sheet No. ......
Measured by .... Date ....
Computed by .... Date ....
Checked by .... Date ....

Requirement	Material	Compass point	Number or length	Width	Height	Volume, area or perimeter	Deductions	Net volume, area or perimeter	Temp. difference	Basic factor	Correction factor	Total B.t.u. per hr. requirement	Total B.t.u. per hr. requirement	Heating surface remarks
Room No. 1														
Infiltration	Window	N		2′ 4″	6′ 6″	20		20	70	145		2900		Basic radiator eff'y. 238 — 3% for length = 231 actual eff'y. 19327 ÷ 231 = 84 sq. ft. required. Installed one window radiator 17 sec. 20-in. high = 85 sq. ft.
Infiltration	Window	N45°	2	2′ 4″	6′ 6″	10		40	70	102		4080		
Window	Single glass	N	3	2′ 4″	6′ 6″	46		46	70	75		3450		
Wall	12-In. brick, furred and plastered	N	17′ 6″		9′ 6″	166	46	120	70	15		1800	12230	
Window	Single glass	E	2	1′ 9″	6′ 6″	23		52	70	75				
Window	Single glass	E		4′ 6″	6′ 6″	29						3900		
Wall	12-In brick, furred and plastered	E		12′	9′ 6″	114	52	62	70	15		930	4830	
Wall	12-In. brick, furred and plastered	W		11′ 6″	9′ 6″	109		109	70	15		1635	1635	
Initial heating	Air, sp. ht. .24, density .078		14′	11′	9′ 6″	1463		1663	20	.019		632	632	
Initial heating	Air, sp. ht. .24, density .078		7′	3′	9′ 6″	200							19327	
Room No. 2														
Infiltration	Window	N		4′	6′ 6″	26		26	70	145		3770		**NORTH SIDE** Basic radiator eff'y. 260 — 2% for length = 255 actual eff'y. 6965 + 975 = 7940 ÷ 255 = 31 sq. ft. required.
Window	Single glass	N		4′	6′ 6″	26		26	70	75		1950		
Wall	12-In brick, furred and plastered	N		11′ 6″	9′ 6″	109	26	83	70	15		1245	6965	
Infiltration	Window	E		6′	6′ 6″	32		32	70	73		2336		Installed one radiator 2 col. 14 sec. 23-in. high = $32\frac{2}{3}$ sq. ft.
Infiltration	Window	E45°	2	2′	6′ 6″	38		38	70	52		1976		
Window	Single glass	E		6′	6′ 6″	39								**EAST SIDE** Basic radiator eff'y. 232 — 20% for enclosure — 3% for length = 180 actual eff'y. 11497 + 974 = 12471 ÷ 180 = 70 sq. ft. required. Installed one radiator 4 col. 18 sec. 22-in. high = 72 sq. ft.
Window	Single glass	E	2	2′	6′ 6″	26		65	70	75		4875		
Wall	12-In. brick, furred and plastered	E		23′	9′ 6″	219	65	154	70	15		2310	11497	
Initial heating	Air, sp. ht. .24, density .078		25′ 6″	20′	9′ 6″	1845		5130	20	.019		1949	1949	
Initial heating	Air, sp. ht. .24, density .078		.0′	3′	9′ 6″	285							20411	
Room No. 3														
Infiltration	Window	W	4	3′	6′ 6″	100		100	70	145		14500		Basic radiator eff'y. 260 — $3\frac{1}{2}$% for length = 251 actual eff'y. 22933 ÷ 251 = 92 sq. ft. required. Installed two radiators each 2 col. 20 sec. 23-in. high = $46\frac{2}{3}$ sq. ft.
Window	Single glass	W	4	3′	6′ 6″	78		78	70	75		5850		
Wall	12-In. brick, furred and plastered	W		20′	9′ 6″	190	78	112	70	15		1680		
Initial heating	Air, sp. ht. .24, density .078		20′	12′ 6′	9′ 6″	2375		2375	20	.019		903	22933	
Room No. 4														
Infiltration	Window	E	2	1′ 6″	6′ 6″	35								Basic radiator eff'y. 256 + $4\frac{1}{2}$% for length = 268 actual eff'y. 9295 ÷ 268 = 35 sq. ft. required. Installed two radiators each 1 col. 6 sec. 38-in. high = 18 sq. ft.
Infiltration	Door	E		5′	7′ 6″	25		60	70	73		4380		
Window	Single glass	E	2	1′ 6″	6′ 6″	20		20	70	75		1500		
Door	$1\frac{1}{2}$-In. wood	E		5′	7′ 6″	38		38	70	35		1330		
Wall	12-In. brick, furred and plastered	E		14′	9′ 6″	133	58	75	70	15		1125		
Initial heating	Air, sp. ht. .24, density .078		19′	14′	9′ 6″	2527		2527	20	.019		960	9295	
Room No. 5														
Infiltration	Door	W		5′	7′ 6″	25		25	70	145		3625		Basic radiator eff'y. 266 + $6\frac{1}{2}$% for length = 283 actual eff'y. 6200 ÷ 283 = 22 sq. ft. required. Installed two radiators each 1 col. 33 sec. 32-in. high = $12\frac{1}{2}$ sq. ft.
Door	$1\frac{1}{2}$-In wood	W		5′	7′ 6″	38		38	70	35		1330		
Wall	12-In. brick, furred and plastered	W		6′	9′ 6″	57	38	19	70	15		285		
Initial heating	Air, sp. ht. .24, density .078		19′	14′	9′ 6″	2527		2527	20	.019		960	6200	

Table 5-1. Heat-Requirement Computation Sheet—*Continued*

Requirement	Material	Compass point	Number or length	Width	Height	Volume, area or perimeter	Deductions	Net volume, area or perimeter	Temp. difference	Basic factor	Correction factor	Total B.t.u. per hr. requirement	Total B.t.u. per hr. requirement	Heating surface remarks
Room No. 6														Basic radiator eff'y. 266 + 6½% for length = 283 actual eff'y.
Infiltration	Window	W		1'	6' 6"	16		16	70	45		2320		3349 ÷ 283 = 12 sq. ft. required.
Window	Single glass	W		1'	6' 6"	7		7	70	75		525		Installed one radiator 1 col. 5 sec. 32-in. high = 12 sq. ft.
Wall	12-In. brick, furred and plastered	W		3' 6"	9' 6"	33	7	26	70	15		390		
Initial heating	Air, sp. ht. .24, density .078		9'	3' 6"	9' 6"	299		299	20	.019		114	3349	
Room No. 7														Register 27 in. above indirect surface with air at 120° gives velocity of 367 ft. per min. Use 50% of this or 184 ft. per min.
Infiltration	Window	E		6'	6' 6"	32		32	70	73		2336		
Infiltration	Window	E45°	2	2'	6' 6"	38		38	70	52		1976		
Window	Single glass	E		6'	6' 6"	39								30-in. vento heaters on 4-in. centers at 100 ft. per min. velocity at 70° gives temp. rise of air from zero to 120 deg. fahr.
Window	Single glass	E	2	2'	6' 6"	26		65	70	75		4875		
Wall	12-In. brick, furred and plastered	E		24'	9' 6"	228	6.	163	70	15		2445	11632	Free area = .225 sq. ft. per section.
Infiltration	Window	W	2	2' 6"	6' 6"	42		42	70	145		6090		
Window	Single glass	W	2	2' 6"	6' 6"	33		33	70	75		2475		**EAST SIDE** Requirements 11632 + 1426 = 13058
Wall	12-In. brick, furred and plastered	W		11'	9' 6"	105	33	72	70	15		1080		$\frac{13058}{.24(120-70)60} = 18.2$ lb. of air per min.
Wall	28-In. brick, furred and plastered	W		11'	9' 6"	105		105	70	8		840	10485	$\frac{18.2}{.075} = 244$ cu. ft. per min. at 70 deg. fahr.
Initial heating	Air, sp. ht. .24, density .078		38'	20'	9' 6"	7220							2852	$\frac{244}{.225 \times 100} = 11$ sections required
Initial heating	Air, sp. ht. .24, density .078		10'	3'	9' 6"	285		7505	20	.019		285	24969	$\frac{144 \times 18.2}{.0682 \times 184} = 208$ sq. in. area hot air duct
														208 + 50% = 312 sq. in area register if 66⅔% free area.
														$\frac{144 \times 18.2}{.0864 \times 184} = 165$ sq. in. area cold air duct.
														WEST SIDE—Requirements 10485 + 1426 = 11911.
														$\frac{11911}{.24(120-70)60} = 16.6$ lb. of air per min.
														$\frac{16.6}{.075} = 222$ cu. ft. per min. at 70 deg. fahr.
														$\frac{222}{.225 \times 100} = 10$ sections required
														$\frac{144 \times 16.6}{.0682 \times 184} = 190$ sq. in. area hot air duct
														190 + 50% = 285 sq. in. area register if 66⅔% Free area.
														$\frac{144 \times 16.6}{.0864 \times 184} = 151$ sq. in. area cold duct.

NOTE—Where wind strikes the window at 45 deg., the resultant velocity at right angles to the window was used for estimating air infiltration

Table 5-2. Heat-Requirement Computation Sheet

Job name, Building B .... No.	Outside Temperature, 0°	Wind vel. Miles aer hr. N 15.	Type of windows, Steel section frames with vent. openings.	Sheet No....
Location	Inside " 65°	" " " " " E 7.5		Measured by .... Date ....
Owner				
Architect	Initial " 40°	" " " " " S 7.5	Type of doors, 2-in. Wood Sliding.	Computed by .... Date ....
Engineer	Heating-up Period hrs. 2	" " " " " W 15.		Checked by .... Date ....

Requirement	Material	Compass point	Number or length	Width	Height	Volume, area or perimeter	Deductions	Net volume, area or perimeter	Temp. difference	Basic factor	Correction factor	Total B.t.u. per hr. requirement	Total B.t.u. per hr. requirement	Heating surface remarks
Section A														
Window	Infiltration	W	10	3′ 6″	4′ 0″	150	....	150	65	134	....	20100	......	
Door	Infiltration	W	....	10′	12′	56	....	56	65	250	....	14000	......	Basic radiator eff'y. 295 + 20% for steam temp. + 3% for room temp. = 365 actual eff'y.
Single glass	Window	W	10	7′	8′	560	....	560	68	75	.97	40740	......	99512 + 70950 = 170462 ÷ 365 = 467 sq. ft. required.
2-In. wood	Door	W	....	10′	12′	120	....	120	66	25	.93	2790	......	
12-In. brick, plain	Wall	W	110′	......	15′	1650	943	607	68	22	.97	12953	......	
16-In. concrete pier	Wall	W	7	2′ 6″	15′	263	....	263	68	35	.97	8929	99512	Installed 477 sq. ft. wall radiator in 5 units.
Door	Infiltration	N	....	10′	12′	56	....	56	65	250	....	14000	......	Basic radiator eff'y. 310 + 20% for steam temp + 3% for room temp. = 383 actual eff'y.
2-In. wood	Door	N	....	10′	12′	120	....	120	66	25	.93	2790	......	24472 ÷ 383 = 64 sq. ft. required.
12-In. brick, plain	Wall	N	30′	......	16′	480	120	360	68	22	.97	7682	24472	Installed 70 sq. ft. wall radiator in 1 unit.
Door	Infiltration	S	....	10′	12′	56	....	56	65	125	....	7000	......	
2-In. wood	Door	S	....	10′	12′	120	....	120	66	25	.93	2790	......	
12-In. brick, plain	Wall	S	30′	......	16′	480	120	360	68	22	.97	7682	17472	Basic radiator eff'y. 310 + 20 % for steam temp. + 3% for room temp. = 383
2-In. wood, paper and gravel	Roof	....	110′	30′	......	3300	....	3300	71	15	1.02	50490	......	17472 ÷ 383 = 46 sq. ft. required.
4-In. concrete on cinder fill	Floor	....	110′	30′	......	3300	....	3300	10	20	.12	7920	......	Installed 49 sq. ft. wall radiator in 1 unit.
Air, sp. ht. .24, density .079	Initial heating	....	110′	30′	16′	52800	....	52800	25	.019	.50	12540	70950 212406	
Section B (Same as Section A	Except air infilt	ration)												
Window	Infiltration	E	10	3′ 6″	4′ 0″	150	....	150	65	68	....	10200	......	
Door	Infiltration	E	....	10′	12′	56	....	56	65	125	....	7000	......	
Single glass	Window	E	10	7′	8′	560	....	560	68	75	.97	40740	......	153562 ÷ 365 = 421 sq. ft. required.
2-In. wood	Door	E	....	10′	12′	120	....	120	66	25	.93	2790	......	
12-In. brick, plain	Wall	E	110′	......	15′	1650	943	607	68	22	.97	12953	......	Installed 432 sq. ft. wall radiators in 5 units.
16-In concrete	Wall	E	7	2′ 6″	15′	263	....	263	68	35	.97	8929	82612	
Door	Infiltration	N	....	10′	12′	56	....	56	65	250	1.00	14000	......	24472 ÷ 383 = 64 sq. ft. required.
2-In. wood	Wood	N	....	10′	12′	120	....	120	66	25	.93	2790	......	Installed 70 sq. ft. wall radiator in 1 unit.
12-In. brick, plain	Wall	N	30′	......	16′	480	120	360	68	22	.97	7682	24472	
Door	Infiltration	S	....	10′	12′	56	....	56	65	125	....	7000	......	17472 ÷ 383 = 46 sq. ft. required.
2-In. wood	Door	S	....	10′	12′	120	....	120	66	25	.93	2790	......	Installed 49 sq. ft. wall radiator in 1 unit.
12-In. brick, plain	Wall	S	30′	......	16′	480	120	360	68	22	.97	7682	17472	
2-In. wood, paper and gravel	Roof	....	110′	30′	......	3300	....	3300	71	15	1.02	50490	......	
4-In. concrete on cinder fill	Floor	....	110′	30′	......	3300	....	3300	10	20	.12	7920	......	
Air, sp. ht. .24, density .079	Initial heating	....	110′	30′	16′	52800	....	52800	25	.019	.5	12540	70950 195506	

Table 5-2. Heat-Requirement Computation Sheet—*Continued*

Requirement	Material	Compass point	Number or length	Width	Height	Volume, area or perimeter	Deductions	Net volume, area or perimeter	Temp. difference	Basic factor	Correction factor	Total B.t.u. per hr. requirement	Total B.t.u. per hr. requirement	Heating surface remarks
Section C lower														
Window	Infiltration	N	2	3′ 6″	4′	30		30	65	134		4020		Basic radiator eff'y. 295 + 20% for steam temp + 3% for room temp. = 365 actual eff'y.
Door	Infiltration	N		10′	12′	56		56	65	250		14000		
Single glass	Window	N	2	7′	8′	112		112	68	75	.97	8148		38518 ÷ 365 = 106 sq ft. required.
2-In. wood	Door	N		10′	12′	120		120	66	25	.93	2790		
12-In. brick, plain	Wall	N		40′	17′	680	232	448	68	22	.97	9560	38518	Installed 108 sq. ft. wall radiators in 2 units.
Window	Infiltration	S	2	3′ 6″	4′	30		30	65	68		2040		Basic radiator eff'y. 295 + 20% for steam temp. + 3% for room temp. = 365 actual eff'y.
Door	Infiltration	S		10′	12′	56		56	65	125		7000		
Single glass	Window	S	2	7′	8′	112		112	68	75	.97	8148		29538 ÷ 365 = 81 sq. ft. required.
2-In. wood	Door	S		10′	12′	120		120	66	25	.93	2790		
12-In. brick, plain	Wall	S		40′	17′	680	232	448	68	22	.97	9560	29538	Installed 90 sq. ft. wall radiators in 2 units.
Section C upper														
Window	Infiltration	W	18	5′	5′ 6″	378		378	65	134		50652		Basic coil eff'y. 360 + 20% for steam temp. + 3% for room temp. = 445 actual eff'y.
Single glass	Window	W	18	5′	5′ 6″	495		495	67	75	.95	35269		
12-In. brick, plain	Wall	W	110′		10′	1100	495	605	65	22	.91	12112		
12-In. brick, plain	Wall	N	40′		10′ 6″	420		420	65	22	.91	8408		262908 ÷ 445 = 591 sq. ft. required.
Single glass	Window	E	18	5′	5′ 6″	495		495	67	75	.95	35269		
12-In. brick, plain	Wall	E	110′		10′	1100	495	605	65	22	.91	12112		
12-In. brick, plain	Wall	S	40′		10′ 6″	420		420	65	22	.91	8408		Installed 604 sq. ft. 1¼-in. pipe coils in 4 units.
2-In. wood, paper and gravel	Roof		110′	40′		4400		4400	66	15	.93	61380		
4-In. concrete on cinder fill	Floor		110′	40′		4400		4400	10	20	.12	10560		
Air, sp. ht. .24, density .079	Initial heating		110′	40′	27′ 6″	121000		121000	25	.019	.5	28738	262908	

CHAPTER VI

Method of Computing and Selecting Heating Surface

DETERMINATION of the heating surface depends first upon the total hourly heat requirements which are assumed to have been calculated as described in the preceding chapter. The heating surface must supply heat units to equal the requirements and should be of the form that best fits the conditions for the room or enclosure.

The method of heat supply must first be determined—that is, whether the heating surface is to be direct, indirect or direct-indirect. The last two methods are used principally where ventilation must be considered in addition to the heating requirements, although the indirect method is considerably used where it is not desired to have the surface located in the room to be heated.

Normally, the heat should be supplied at the locations where the greatest requirements occur, and this is generally at the windows, where, in addition to a high transmission requirement, there is the air infiltration requirement as well.

Rooms or enclosures where more than one unit of radiation is to be installed should have the heating surface divided in proportion to the requirements of the spaces served.

Heating surface placed under the windows should not project above the sills, should be as wide as the window openings, and should also be installed with a 2½-inch space between the wall and the surface, as this distance gives maximum efficiency of heat emission.

Direct heating surface, inasmuch as it is used in a large majority of installations, should be considered first. Residences, office, school, library, hospital and similar buildings usually have cast-iron column radiation

Fig. 6-1. Cast-iron wall radiation on side walls under windows, for heating a factory building

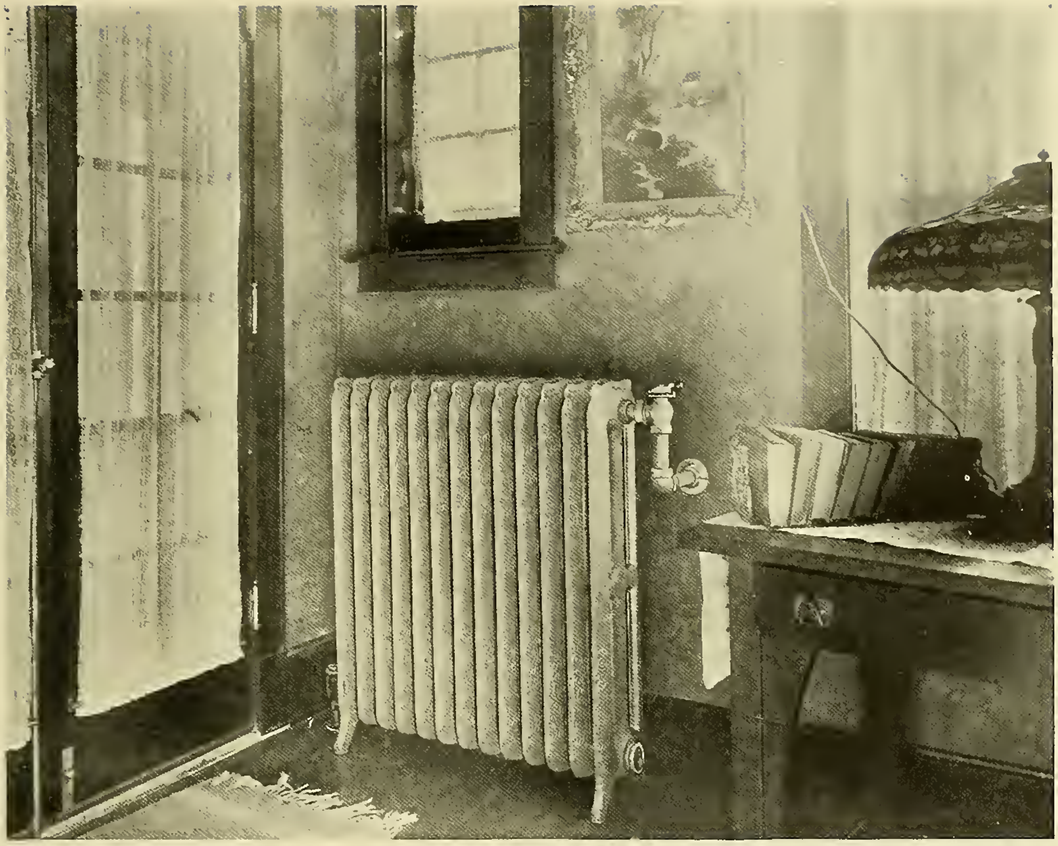


Fig. 6-2. Connections to a direct hot-water type radiator showing modulation supply valve and thermostatically actuated return trap

together with some cast-iron wall radiation. Factory and manufacturing buildings are usually heated by means of wrought-iron or steel pipe coils or cast-iron wall radiation.

Hot-water pattern radiation is preferable for those systems in which modulation supply valves are to be used. The supply valve should be placed at the upper inlet and the return trap at the lower opening diagonally opposite.

Good practice in the use of groups of wall radiation suggests that no individual group exceed 30 ft. in length, as expansion and contraction become an important factor on longer groups. Where greater lengths of this type of radiation must be used, the supply connection should be made at top and bottom and expansion and contraction properly provided for.

Pipe coil practice demands a spring or mitre piece in the coil to provide for expansion and contraction, and the desirable length is limited to 60 ft. not including the mitre piece. Coils should be securely anchored at the return header so as to throw the expansion toward the mitre end, the length of which should be not less than one-twelfth the coil length for 1-in. pipe and one-tenth for $1\frac{1}{4}$-in. or $1\frac{1}{2}$-in. pipe.

Fig. 6-3. Arrangement of cast-iron wall radiation on side wall of a factory building

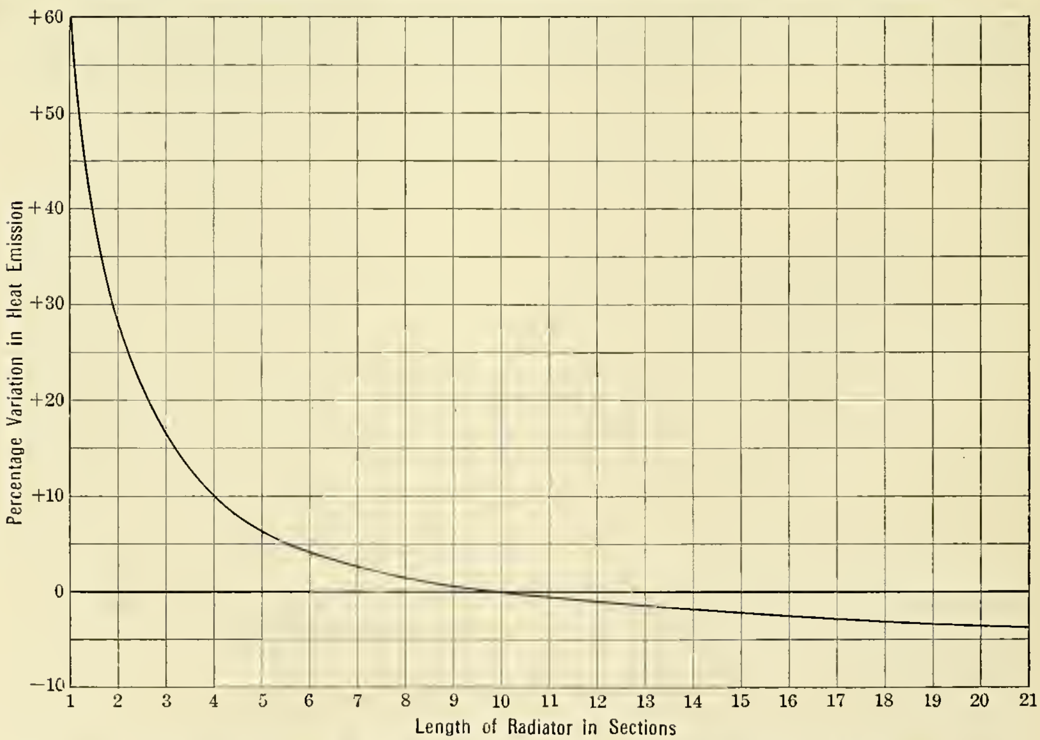


Fig. 6-4. Percentage of variation in heat emitted from cast-iron heating surface per square foot for various numbers of sections as compared with a standard 10-section radiator

The amount of heat emitted from any given type of direct heating surface is usually stated in B.t.u. per hour per square foot of heating surface. This heat is given off in two ways, by convection directly to the air which passes over the heated surface, and by radiation directly to surrounding materials independent of that carried off by the air. The heat given off by radiation does not heat the air through which it passes, but travels in straight lines and heats the objects upon which it impinges.

After selecting the type of surface best suited for the particular case, the number of square feet of heating surface required should be determined next. The total number of heat units that must be supplied per hour divided by the heat units emitted per hour per square foot of heating surface gives the required surface in square feet.

Table 6-1 will be of assistance in determining the heat emitted by different types of surface.

Table 6-1. B.t.u. Emitted per Hour per Square Foot of Heating Surface*
Radiators 10 Sections Long
Steam Temperature, 215 deg. fahr. Room Temperature, 70 deg. fahr.

Number of columns	Height of radiator	B.t.u. by convection	B.t.u. by radiation	Total B.t.u.	Percent convected heat of total heat
One	38 in.	150	106	256	58.6
"	32 in.	158	108	266	59.4
"	26 in.	162	111	273	59.4
"	23 in.	160	119	279	57.4
"	20 in.	166	117	283	58.7
Two	45 in.	148	86	234	63.
"	38 in.	148	92	240	62.
"	32 in.	154	94	248	62.
"	26 in.	149	106	255	58.
"	23 in.	151	109	260	58.
"	20 in.	153	112	265	58.
Three	45 in.	142	76	218	65.
"	38 in.	147	79	226	65.
"	32 in.	158	75	233	68.
"	26 in.	166	75	241	69.
"	22 in.	166	82	248	67.
"	18 in.	162	92	254	64.
Four	45 in.	149	56	205	73.
"	38 in.	150	60	210	71.5
"	32 in.	151	66	217	69.5
"	26 in.	155	70	225	69.
"	22 in.	156	76	232	67.
"	18 in.	151	87	238	63.5
Wall radiation					
3 in. wide	14 in.	152	171	323	47.
" "	22 in.	154	156	310	49.7
" "	29 in.	138	157	295	48.
Pipe coil	6-1¼ in. pipes			360	
" "	8-1¼ in. "			343	
" "	10-1¼ in. "			330	
" "	12-1¼ in. "			319	

* John R. Allen, *A. S. H. & V. E. Journal*—January, 1920

From Table 6-1 it will be noted that low, narrow surface is most efficient and that the efficiency decreases as the height and width increase.

Some other factors and their effect upon the efficiency of the radiating surface are worthy of explanation.

The preceding table is based upon a radiator 10 sections long. As the number of sections decreases, the efficiency increases, due to increase of the more efficient end-section surface in proportion to total heating surface; also a short radiator emits proportionally more radiant heat than a longer one. Figure 6-4 shows the effect of varying the number of sections, and that increasing the number of sections above 10 has not as much effect as decreasing the number below 10. It will also be noted that a 4-section radiator will give off about 10 per cent more heat per square foot of surface than one 10 sections long.

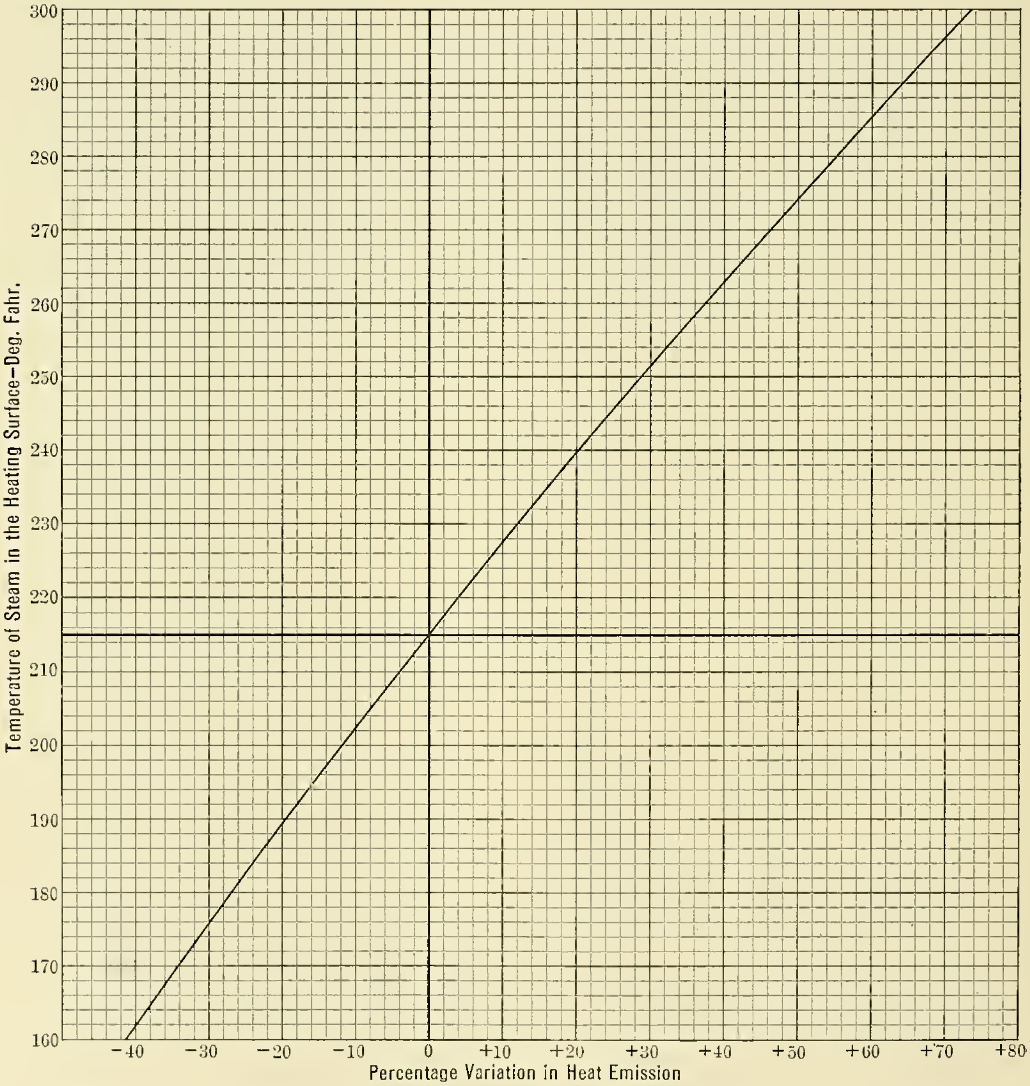


Fig. 6-5. Percentage variation in heat emitted from heating snrface due to varying the steam temperature from 215 deg. fahr., room temperature 70 deg. fahr.

Where 215 deg. fahr. is considered as the standard temperature of steam in the heating surface, the effect upon the heat emission of the surface due to varying this temperature is shown in Figure 6-5. The percentage variation can be read directly from the curve.

Example: If steam at a temperature of 230 deg. fahr. is supplied to the radiator, the heat emission will be increased 12 per cent over one supplied with steam at 215 deg. fahr.

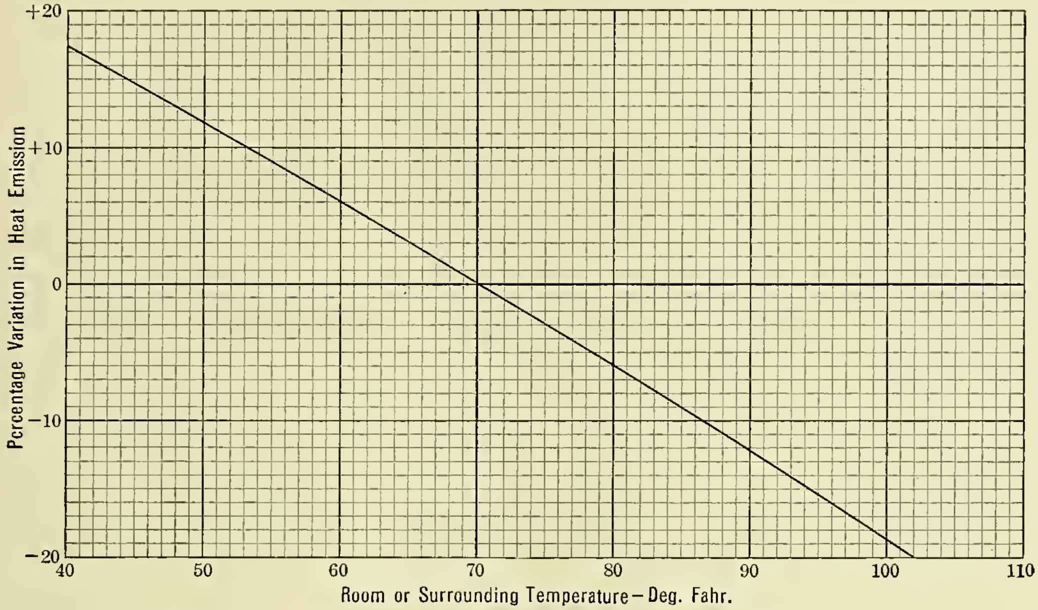


Fig. 6-6. Percentage variation in heat emitted from heating surface due to varying the room temperature from 70 deg. fahr.

The surrounding or room temperature is taken at 70 deg. fahr. as a standard. The effect upon the heat emitted from heating surface, due to varying this temperature, is shown graphically in Figure 6-6. From the curve it will be observed that, for instance, a radiator in a room temperature of 60 deg. fahr. will emit 6 per cent more heat than the same radiator in a room temperature of 70 deg. fahr.

The effect on heat emission due to variation in steam temperature is much greater than an equal temperature variation in the surrounding or room temperature.

The following example will illustrate the use of the curves in Figures 6-4, 6-5 and 6-6 for determining the heat emission under given conditions; it is desired to know the B.t.u. emitted per hour per square foot of heating surface of a standard cast-iron radiator, two columns wide, 38 in. high, and six sections long when supplied with steam at 240 deg. fahr. and located in a room heated to 80 deg. fahr.

Referring to Table 6-1, a similar radiator except that it is 10 sections long, gives off 240 B.t.u. per hr. per sq. ft., with steam at 215 deg. fahr. in room temperature 70 deg. fahr. A radiator six sections long is 4.5 per cent. more efficient (Figure 6-4), when supplied with steam at 240 deg.

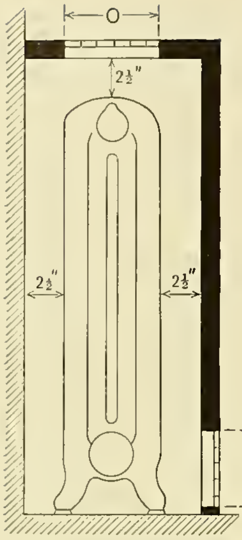


Fig. 6-7

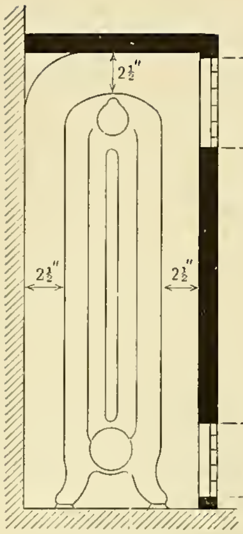


Fig. 6-8

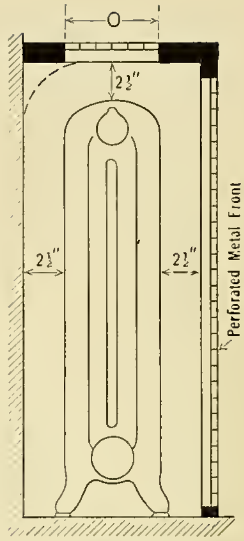


Fig. 6-9

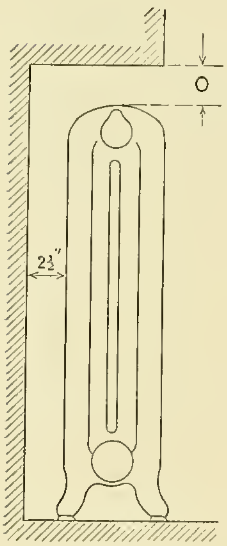


Fig. 6-10

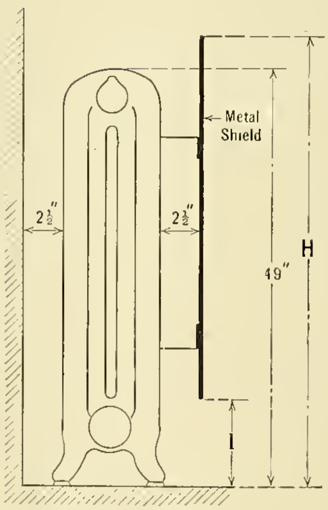


Fig. 6-11

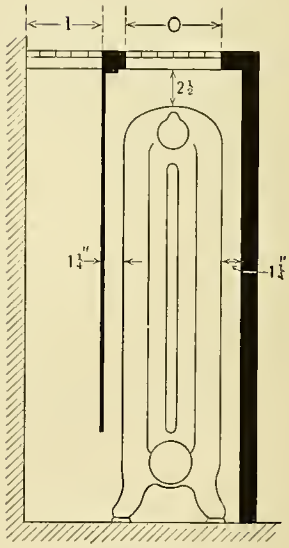


Fig. 6-12

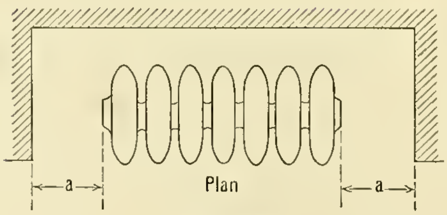


Fig. 6-13

Length of all outlets O = length of radiator

Length of all inlets I = length of radiator

Width of all outlets O = width of radiator or as given in table

Screens or grilles have 44 per cent free area

Enclosures for radiators

fahr., the efficiency is increased 20 per cent (Figure 6-5), and if located in a room heated to 80 deg. fahr. there is a decrease in efficiency of 6 per cent (Figure 6-6). The heat emission of the radiator required would be 240 x 1.045 x 1.20 x 0.94=283 B.t.u. per hr. per sq. ft. of radiating surface.

Painting a radiator influences only the heat emitted by radiation, the convection factor remaining practically unchanged. As paint affects the surface only, the number of coats makes little difference. It seems to depend on the last coat applied and when made of flake metal the result is more marked.

Direct radiators are sometimes set behind grilles or screens, in window enclosures or wall recesses, all of which greatly decrease the efficiency of the radiation.

Tests by Professor Brabbee, as reported by George Stumpf, Heating and Ventilating Magazine, May, 1914, show that a radiator in an enclosure is most efficient when located with 2½ inches between the wall and radiator and between the inside of the enclosure and the radiator. Abstracts from these tests follow.

The inlet and outlet openings of any form of enclosure should extend at least the entire length of the radiator. The width of the outlet is usually made that of the radiator. Tests show little gain in efficiency for wider outlets, but a decrease of about 5 per cent for each inch narrower than that of the radiator.

The outlets and inlets in Tables 6-7 to 6-13 are the full length of the radiators. The width of outlet O is the width of the radiator except in Table 6-4, where it is as given. The width of inlet I is as stated in the tables.

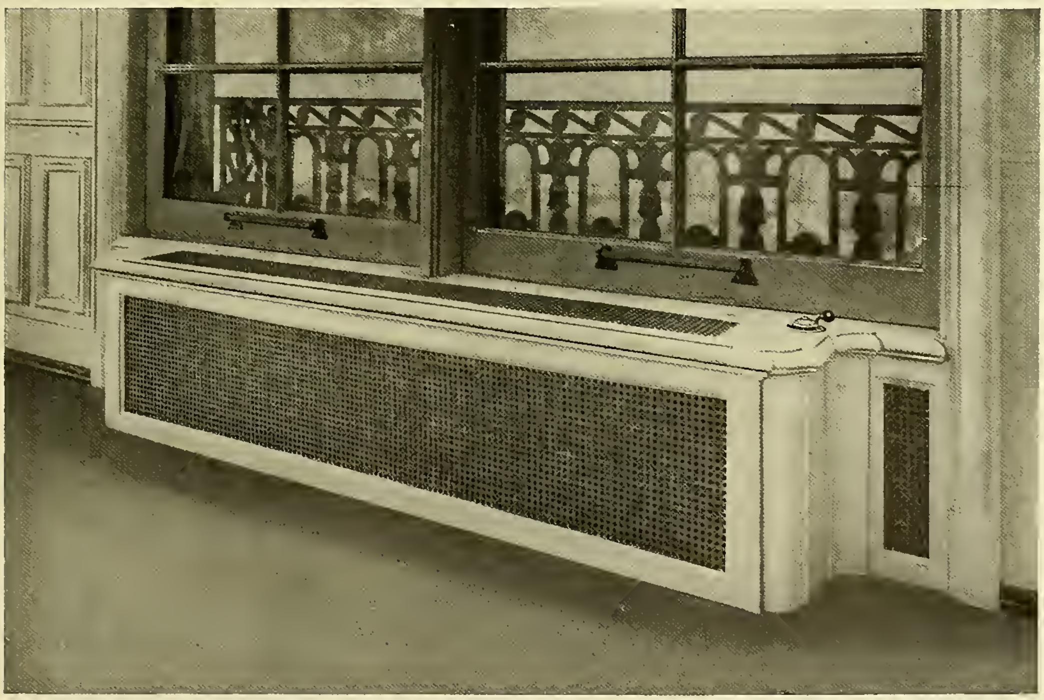


Fig. 6-14. An enclosed radiator having grilles or screens on front and top of enclosure. The modulation supply valve control is shown on top of enclosure

Both openings are covered with screen of 44 per cent free area.

The design of the screen or grille has no effect provided the free area is not changed.

Figure 6-7 shows a form of enclosure frequently used.

Table 6-2. Decrease in Radiator Efficiency with Form of Enclosure Shown in Fig. 6-7

Radiator width	Radiator height	Width of I	Decrease in efficiency
Two-column	42 in. and over	9 in.	15%
" "	Under 42 in.	9 in.	20%
" "	Under 42 in.	5 in.	25%
Three-column	42 in. and over	9 in.	15%
" "	32 in. to 38 in.	9 in.	15%
" "	32 in. to 38 in.	7 in.	20%
" "	26 in. and under	9 in.	20%
" "	26 in. and under	5 in.	25%

If the width of inlet is made equal to the free area and not screened, the efficiency reduction will remain as above.

Another form of enclosure, Figure 6-8, gives the effect upon the radiation efficiency as shown in Table 6-3.

Table 6-3. Decrease in Radiator Efficiency with Form of Enclosure Shown in Fig. 6-8

Radiator width	Radiator height	Width of O	Width of I	Decrease in efficiency
Two-column	42 in. and over	8 in.	8 in.	20%
" "	32 in. to 38 in.	9 in.	9 in.	20%
" "	32 in. to 38 in.	7 in.	7 in.	25%
" "	26 in. and under	6 in.	6 in.	33%
Three-column	26 in. and over	9 in.	9 in.	20%
" "	26 in. and over	6 in.	6 in.	25%

Enclosure of the form shown in Figure 6-9 is sometimes used and by test gives the following effect:

Table 6-4. Decrease in Radiator Efficiency with Form of Enclosure Shown in Fig. 6-9

Perforated screen full front of enclosure only—decrease in efficiency	20%
Same screen with deflector— " " "	15%

If an outlet O is provided in addition to front screen and made equal to width and length of the radiator, the efficiency decreases only 10 per cent.

Sometimes it is desirable to set the radiators in wall recesses, as shown in Figures 6-10 and 6-13 which causes a decrease in efficiency as follows:

Table 6-5. Decrease in Radiator Efficiency Due to Wall Recess Fig. 6-10

When 0 = 1½ inches—decrease in efficiency	11%
" 0 = 3 " " " "	7.3%
" 0 = 4 " " " "	6%

The distance a has little or no effect, and therefore need only be sufficient for connections to the radiator.

A shield in front of a radiator as shown in Figure 6-11 increases the radiator efficiency as follows:

Fig. 6-15. An enclosed radiator in a window seat, with grilles of rattan cane. The modulation supply valve control is placed on the window seat

Table 6-6. Increase in Radiator Efficiency by Use of a Shield Fig. 6-11

Height of shield, H	52 in.	52 in.	52 in.	72 in.
Width of open slot, 1	$6\frac{1}{2}$ in.	9 in.	12 in.	12 in.
Increase in efficiency	2.2%	6.3%	12.5%	13%

Another form of enclosure, shown in Figure 6-12, by test gives the following effect upon the radiator efficiency:

Table 6-7. Decrease in Radiator Efficiency with Form of Enclosure Shown in Fig. 6-12

Width 1	8 in.	6 in.	5 in.	4 in.	3 in.
Decrease in efficiency	10%	15%	20%	25%	33%

Table 6-8. Comparative B.t.u. Transmission and Cost of Cast-Iron Heating Surface

Based on 3-column 30-in. radiation as 1.00

Rad. height	Relative cost of radiator per sq. ft.				B.t.u. given off per sq. ft.				Relative cost based on heating efficiency			
	1 Column	2 Columns	3 Columns	4 Columns	1 Col.	2 Col's	3 Col's	4 Col's	1 Column	2 Columns	3 Columns	4 Columns
18″			1.43	1.43			254	238			1.27	1.36
20″	1.49	1.43			283	265			1.19	1.22		
22″			1.28	1.28			248	232			1.17	1.25
23″	1.38	1.31			279	260			1.10	1.14		
26″	1.30	1.25	1.18	1.18	273	255	241	225	1.08	1.11	1.10	1.18
32″	1.18	1.13	1.08	1.08	266	248	233	217	1.01	1.03	1.04	1.12
38″	1.09	1.04	1.00	1.00	256	240	226	210	.96	.95	1.00	1.08
45″		1.04	1.00	1.00		234	218	205		1.01	1.04	1.11

These tables are based on investigations of 10-section radiators

For Radiators under 6-section, the B.t.u. per sq. ft. increases rapidly and the tables cannot be used with accuracy. Above 6-section the error is small

Table 6-8 will be of interest as it compares the relative costs of cast-iron heating surface of different heights and number of columns where the efficiency of the surface is taken into consideration.

As an example, compare the relative cost of 3-column 38-in. with single-column 23-in. surface. The 3-column surface cost is figured as 1.00 and it emits 226 B.t.u. per sq. ft. per hr. The single-column radiator cost is 1.36 but it emits 279 B.t.u. per sq. ft. per hr. Although the actual cost per square foot for the single-column radiator is 36 per cent more than for the 3-column, the 1-column radiator is 23 per cent more efficient in heat emission. If this increase in heating efficiency is considered, the cost of the single-column radiation is only 10 per cent more.

Indirect heating surface generally refers to that located below and outside of the room to be heated. (See Figure 6-16.) The heat is delivered to the room by a system of ducts that convey fresh air from outside. The air passes over the surface, is heated and then discharged into the room through register faces located in the room floor or wall. This method of heating is

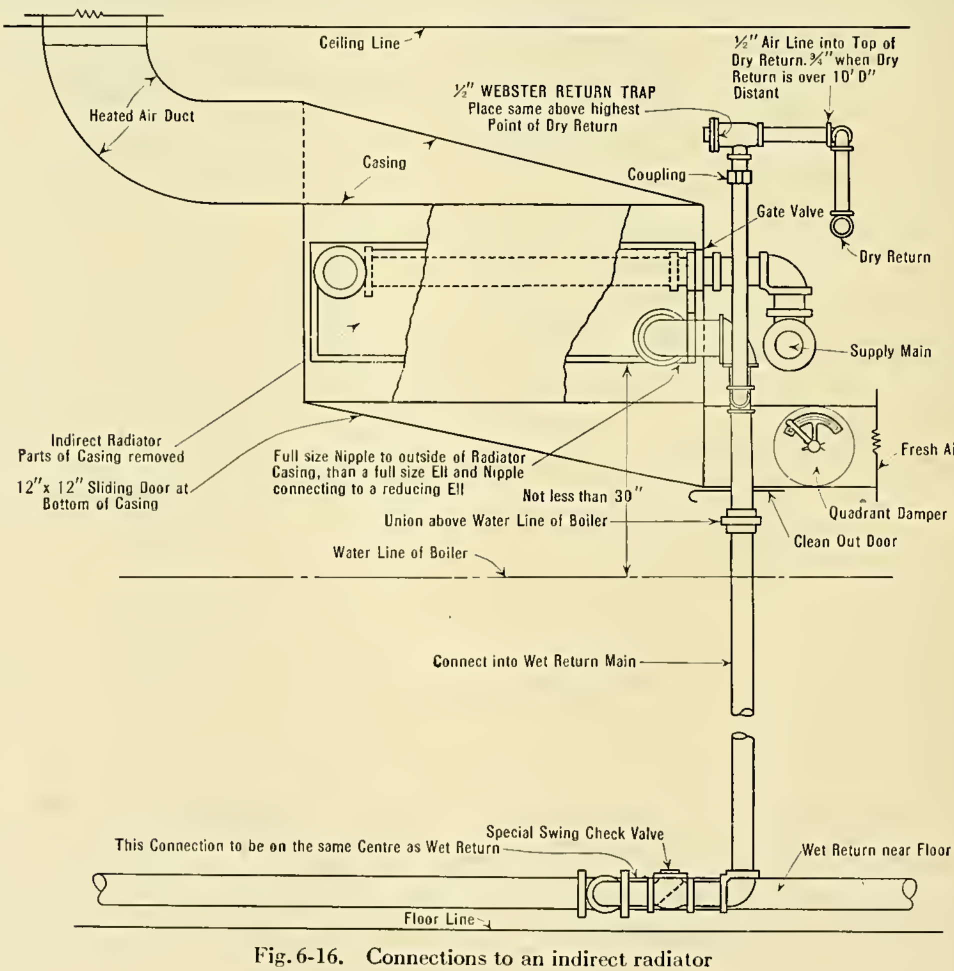


Fig. 6-16. Connections to an indirect radiator

called *fresh air indirect*, as a constant supply of fresh heated air is delivered into the room. The cold air duct is sometimes so arranged that outside air may be closed off and air taken from the basement in extreme cold weather.

Where the air supply is taken from the room, passed over the heating surface and then discharged into the room again, the method is known as *recirculating indirect.*

In either system no heating surface is located in the room to be heated.

The indirect method of heating is most used in the principal rooms of residences, clubs, churches and similar types of buildings, and is much more expensive to install and to operate than is the direct system.

All rooms heated by the fresh air indirect system must be provided with vents for the escape of the air replaced by that delivered by the "indirect stack," as this type of heating surface is often called.

Many variable factors, each of prime importance, enter into an accurate calculation of the proper proportions of a system of this type. These variables include velocity and direction of the wind, frictional resistance to the air flow in the ducts, and the loss of heat due to transmission through the walls of hot-air ducts.

Each manufacturer of heating surface for this system has his own special design, which is usually sold by catalogue ratings in square feet of surface. Reliable data as to the free area between sections and the heating effect under the variable conditions of steam and air temperatures at various air velocities are unfortunately not available for each make of heating surface used in this method of heating. Proper values are very difficult to assign to the variable factors, and the several rules for determining the proper proportions of such a system are all based upon some standard conditions and assumptions.

The general principle of an indirect system is the delivery of air to the room at a temperature higher than that of the room, and in such volume that in cooling to room temperature, sufficient heat units are given up to replace those required for transmission, infiltration and other requirements.

The requirements for this method of heating are usually computed in the following way:

First: Calculate the total heat requirements in B.t.u. per hour for the room to be heated as described in Chapter 5.

Second: Determine the height of the column of heated air; that is, the distance from center of indirect stack to center of the room register.

Third: Assume the temperature of the air entering the room. This is usually taken about 120 deg. fahr. where air enters the radiator at zero and the radiator is supplied with steam at atmospheric pressure or slightly above.

Fourth: Determine the velocity of air due to difference in densities between heated and outside air for columns of equal height.

Fifth: Ascertain from the manufacturer of the selected type of heating surface the velocity at which air must pass through the surface to produce the final required temperature, when the surface is supplied with steam at a predetermined temperature and air enters the heating stack at the minimum outside temperature. Ascertain also the temperature of the air on which this performance is based, the free area between the sections, and the

number of square feet of heating surface per section.

The amount of heating surface may then be determined as follows:

H = total B.t.u. losses per hour for the room.

t_1 = temperature of air entering the room.

t_2 = temperature of air in room (room temperature).

t_3 = temperature of air on which heating surface performance is based.

d = density of air at temperature t_3.

v = performance velocity of air in feet per minute.

a = free area per section of heating surface in square feet.

$$\frac{H}{.2375\ (t_1 - t_2)\ 60} = \text{pounds of air required per minute} = P$$

where 0.2375 is the specific heat of the air.

$\frac{P}{d}$ = cubic feet of air per minute at t_3.

$\frac{P}{d} \div av$ = number of sections of heating surface required from which the square feet of heating surface can be determined.

The sizes of the ducts or flues for conveying the air to and from the heating surface are dependent upon the velocity of the air due to the unbalanced air column. This velocity may be determined theoretically from the formula:

$$v = 480\sqrt{\frac{h\ (t - t_0)}{460 + t_0}}$$

in which

v = velocity in feet per minute.

h = height of warm air column in feet or distance from center of heating surface to center of register.

t = average temperature of air in column.

t_0 = average temperature of outside air.

To allow for friction in ducts, through heating surface, register face and elsewhere, velocities of one-third of the theoretical may be assumed.

The area of the hot-air duct may be determined as follows:

$$\text{Area in square inches} = \frac{144\ P}{d\ v}$$

in which

P = pounds of air required per minute.

d = density of air at average temperature in hot-air duct.

v = velocity in feet per minute in duct.

The register can have a free area equal to the area of the hot-air duct where velocity in hot-air duct is not in excess of 300 ft. per min. For higher velocities the register area should be increased. The area of the cold-air duct can be determined in a manner similar to the hot-air duct area, using density of the air at the cold inlet temperature.

Direct-indirect heating surface, as the name implies, consists of radiators arranged so that a portion of each serves on the indirect principle and the remainder as a direct radiator; the entire surface, however, is located in the

room to be heated. This combination is accomplished by providing a direct radiator and installing a metal box base under some of the sections. Cold fresh air is taken from the outside of the building directly through the wall and connected to this box base. The fresh air passes up through its portion of the surface into the room. The balance of the surface acts as plain direct heating surface.

This method of heating has come into quite general use in recent years in some localities where the state ventilation laws for public buildings specify either the quantity of air to be supplied per minute per person, or the number of square inches of fresh-air inlet duct per person. The latter requirement can be met with this type of heating surface.

The size of the opening in the wall or the wall box determines the size of the box base, and the number of sections of the radiator enclosed by the box base are to be considered as available only for heating the incoming air.

Sufficient additional direct heating surface must be provided, either by adding sections to the radiator, extending same outside of the box base on either end, or by installing separate units for supplying the heat necessary for requirements of the wall, glass and infiltration, as already mentioned.

Vent flues must be extended from all rooms heated and ventilated by this method.

In order to obtain desired air movement and prevent back draft in flues, they must have aspirating radiation or rotary type ventilators.

The radiation best suited for direct-indirect surface is that with high and wide sections. One manufacturer of the most modern devices for this type of system states the size of the ventilating base, together with its capacity, fresh-air inlet area and amount of radiating surface to be enclosed, as given in Table 6-9.

Table 6-9. Data for Direct-indirect Heating Surface Offered by One Manufacturer. *Not Standard for Other Similar Equipment*

Size of wall box	Capacity in cu. ft. per min.	Area of fresh air opening	Heating surface
8 in. x 20 in.	180	120 sq. in.	50 sq. ft.
8 in. x 24 in.	240	144	50
8 in. x 30 in.	300	180	60
10½ in. x 20 in.	270	160	50
10½ in. x 24 in.	330	192	60
10½ in. x 30 in.	420	240	60

As an example of selecting and computing heating surfaces, refer to the heat requirements as shown on Pages 38-39 for the various rooms in Figure 5-1, Page 36, and assume that steam will be used at 215 deg. fahr., or 1-lb. per sq. in. pressure.

Room 3 requires a total of 22933 B.t.u. per hr. and is to be heated by means of direct radiation. The window sills are 24 in. high. Therefore, 23-in. high radiators should be installed. For a room of this size, it appears that 2-column radiation should give sufficient surface. The B.t.u. emitted by 2-column, 23-in. high radiation is given in Table 6-1

Table 6-10. Surface in Square Feet of One to Twelve $1\frac{1}{4}$-inch Pipe Coil, 1 to 100 Feet Long

(For other sizes of pipe, see note at bottom of next page.)

Length of coil in feet	Number of $1\frac{1}{4}''$ pipes											
	1	2	3	4	5	6	7	8	9	10	11	12
	Square feet of heating surface											
1	0.43	0.86	1.29	1.72	2.15	2.58	3.01	3.44	3.87	4.30	4.73	5.16
2	1	2	3	3	4	5	6	7	8	9	9	10
3	1	3	4	5	6	8	9	10	12	13	14	15
4	2	3	5	7	9	10	12	14	15	17	19	21
5	2	4	6	9	11	13	15	17	19	22	24	26
6	3	5	8	10	13	15	18	21	23	26	28	31
7	3	6	9	12	14	18	21	24	27	30	33	36
8	3	7	10	14	17	21	24	28	31	34	38	41
9	4	8	12	15	19	23	27	31	35	39	43	46
10	4	9	13	17	22	26	30	34	39	43	47	52
11	5	9	14	19	24	28	33	38	43	47	52	57
12	5	10	15	21	26	31	36	41	46	52	57	62
13	6	11	17	22	28	34	39	45	50	56	61	67
14	6	12	18	24	30	36	42	48	54	60	66	72
15	6	13	19	26	32	39	45	52	58	65	71	77
16	7	14	21	28	34	41	48	55	62	69	76	83
17	7	15	22	29	37	44	51	58	66	73	80	88
18	8	15	23	31	39	46	54	62	70	77	85	93
19	8	16	25	33	41	49	57	65	74	82	90	98
20	9	17	26	34	43	52	60	69	77	86	95	103
21	9	18	27	36	45	54	63	72	81	90	99	108
22	9	19	28	38	47	57	66	76	85	95	104	114
23	10	20	30	40	49	59	69	79	89	99	109	119
24	10	21	31	41	52	62	72	83	93	103	114	124
25	11	22	32	43	54	65	75	86	97	108	118	129
26	11	22	34	45	56	67	78	89	101	112	123	134
27	12	23	35	46	58	70	81	93	104	116	128	139
28	12	24	36	48	60	72	84	96	108	120	132	144
29	12	25	37	50	62	75	87	100	112	125	137	150
30	13	26	39	52	65	77	90	103	116	129	142	155
31	13	27	40	53	67	80	93	107	120	133	147	160
32	14	28	41	55	69	83	96	110	124	138	151	165
33	14	28	43	57	71	85	99	114	128	142	156	170
34	15	29	44	58	73	88	102	117	132	146	161	175
35	15	30	45	60	75	90	105	120	135	151	166	181
36	15	31	46	62	77	93	108	124	139	155	170	186
37	16	32	48	64	80	95	111	127	143	159	175	191
38	16	33	49	65	82	98	114	131	147	163	180	196
39	17	34	50	67	84	101	117	134	151	168	184	201
40	17	34	52	69	86	103	120	138	155	172	189	206
41	18	35	53	71	88	106	123	141	159	176	194	212
42	18	36	54	72	90	108	126	144	163	181	199	217
43	18	37	55	74	92	111	129	148	166	185	203	222
44	19	38	57	76	95	114	132	151	170	189	208	227
45	19	39	58	77	97	116	135	155	174	194	213	232
46	20	40	59	79	99	119	138	158	178	198	218	237
47	20	40	61	81	101	121	141	162	182	202	222	243
48	21	41	62	83	103	124	144	165	186	206	227	248
49	21	42	63	84	105	126	147	169	190	211	232	253
50	22	43	65	86	108	129	151	172	194	215	237	258

Table 6-10. Surface in Square Feet of One to Twelve 1¼-inch Pipe Coil, 1 to 100 Feet Long—*Continued*

Length of coil in feet	Number of 1¼" pipes											
	1	2	3	4	5	6	7	8	9	10	11	12
	Square feet of heating surface											
51	22	44	66	88	110	132	154	175	197	219	241	263
52	22	45	67	89	112	134	157	179	201	224	246	268
53	23	46	68	91	114	137	160	182	205	228	251	273
54	23	46	70	93	116	139	163	186	209	232	255	279
55	24	47	71	95	118	142	166	189	213	237	260	284
56	24	48	72	96	120	144	169	193	217	241	265	289
57	25	49	74	98	123	147	172	196	221	245	270	294
58	25	50	75	100	125	150	175	200	224	249	274	299
59	25	51	76	101	127	152	178	203	228	254	279	304
60	26	52	77	103	129	155	181	206	232	258	284	310
61	26	52	79	105	131	157	184	210	236	262	289	315
62	27	53	80	107	133	160	187	213	240	267	293	320
63	27	54	81	108	135	163	190	217	244	271	298	325
64	28	55	83	110	138	165	193	220	248	275	303	330
65	28	56	84	112	140	168	196	224	252	280	307	335
66	28	57	85	114	142	170	199	227	255	284	312	341
67	29	58	86	115	144	173	202	230	259	288	317	346
68	29	58	88	117	146	175	205	234	263	292	322	351
69	30	59	89	119	148	178	208	237	267	297	326	356
70	30	60	90	120	151	181	211	241	271	301	331	361
71	31	61	92	122	153	183	214	244	275	305	336	366
72	31	62	93	124	155	186	217	248	279	310	341	372
73	31	63	94	126	157	188	220	251	283	314	345	377
74	32	64	95	127	159	191	223	255	286	318	350	382
75	32	65	97	129	161	194	226	258	290	323	355	387
76	33	65	98	131	163	196	229	261	294	327	359	392
77	33	66	99	132	166	199	232	265	298	331	364	397
78	34	67	101	134	168	201	235	268	302	335	369	402
79	34	68	102	136	170	204	238	272	306	340	374	408
80	34	69	103	138	172	206	241	275	310	344	378	413
81	35	70	104	139	174	209	244	279	313	348	383	418
82	35	71	106	141	176	212	247	282	317	353	388	423
83	36	71	107	143	178	214	250	286	321	357	393	428
84	36	72	108	144	181	217	253	289	325	361	397	433
85	37	73	110	146	183	219	256	292	329	366	402	439
86	37	74	111	148	185	222	259	296	333	370	407	444
87	37	75	112	150	187	224	262	299	337	374	412	449
88	38	76	114	151	189	227	265	303	341	378	416	454
89	38	77	115	153	191	230	268	306	344	383	421	459
90	39	77	116	155	194	232	271	310	348	387	426	464
91	39	78	117	157	196	235	274	313	352	391	430	470
92	40	79	119	158	198	237	277	316	356	396	435	475
93	40	80	120	160	200	240	280	320	360	400	440	480
94	40	81	121	162	202	243	283	323	364	404	445	485
95	41	82	123	163	204	245	286	327	368	409	449	490
96	41	83	124	165	206	248	289	330	372	413	454	495
97	42	83	125	167	209	250	292	334	375	417	459	501
98	42	84	126	169	211	253	295	337	379	421	464	506
99	43	85	128	170	213	255	298	341	383	426	468	511
100	43	86	129	172	215	258	301	344	387	430	473	516

Note: For all practical purposes, figure 1-3 sq. ft. of outside surface per lineal foot of 1-in. pipe; and 1-2 sq. ft. for 1 1-2 in. pipe

as 260 B.t.u. per hr. per sq. ft. of surface. As these radiators will be 20 sections long instead of the standard 10, on which the above efficiency was based, the efficiency, or B.t.u. emitted will be reduced by 3.5 per cent, making an actual efficiency of 251. This divided into the total heat requirements gives 91 sq. ft. of heating surface required, which is supplied by two units of 46⅔ sq. ft. each as marked on the plan.

Data as above for determination of the other units are marked on the plan. Room 7, which is to be heated by indirect surface, is calculated as follows: The total requirements for the east side are 13058 B.t.u. per hr., and assuming that the air enters the room at 120 deg. fahr., the pounds of air required in accordance with formula on Page 54 would be 18.2 per minute.

Vento radiation 30 inches long on 4-in. centers gives a temperature rise of air from zero to 120 deg. fahr. at 100 ft. per min. velocity, measured at 70 deg. fahr. volume. The free area per section is 0.225 sq. ft.

The pounds of air as found above divided by the density at 70 deg. fahr., or 0.0749, gives 244 cu. ft. of air per minute.

This volume divided by the velocity, then by the free area per section, gives eleven sections required.

The distance from the center of the radiation to the floor above is 27 inches, which head with 120 deg. fahr. temperature difference gives a theoretical velocity of 367 ft. per min., by the formulæ on Page 54. For determining the size of the ducts, one-half of this value, or 184 ft. per min. velocity may be used.

Using formula on Page 54 with a density for air at 120 deg. fahr., the area of the hot-air duct is 208 sq. in. The register if of 66⅔ per cent free area should contain 312 sq. in.

The cold-air duct by the above formula, using air density at zero, should have a sectional area of 165 sq. in.

The indirect surface for the requirement of the west side of this room was calculated similarly.

As another example, to determine the radiation necessary to supply the heat required for the factory building as calculated in the previous chapter and shown in Figure 5-2, Page 37.

Assume that steam at 10-lb. per sq. in. pressure or at a temperature of 240 deg. fahr. is available for heating this building under maximum load conditions. The increase in B.t.u. emission of the heating surfaces for this increased temperature above the standard or basic temperature is 20 per cent, and there would be a further increase in efficiency of 3 per cent due to a 65-deg. fahr. instead of 70-deg. fahr. room temperature.

This would make a total increase of 23.6 per cent in B.t.u. emitted per hour per sq. ft. of heating surface for this installation, over the basic value.

The monitor portion of the building is provided with 1¼-in. pipe coils under the windows, with expansion springs at the ends, as shown. For the lower portion of the building cast-iron wall surface is to be installed as shown. The efficiency of the heating surface and method of determining the amount of surface are shown on the plan.

CHAPTER VII

Ventilation Problems as They Affect the Design of Heating Systems

VENTILATION in the past was based on more or less traditional and unscientific standards, but is now receiving more of the consideration warranted by its importance.

The necessity of providing adequate ventilating facilities for public buildings and buildings for various classes of industrial operations has been recognized by the legislative bodies of numerous states and cities, which have passed laws and ordinances governing the quantity of air to be supplied per person, and in some instances also the locations from which the air supply is to be brought into the room and the vitiated air removed.

Ventilation is classed, and rightly so, as a branch of applied science, and it is the duty of the ventilating engineer to apply the principles of this science to the problems with which he is dealing in such a manner that the results obtained will produce the most healthful and comfortable conditions in the ventilated rooms.

A ventilating system may be very satisfactory in regard to the quantity and means of distribution of the air but still fail to produce healthful and comfortable conditions. A good ventilating system should produce immediate physical comfort. The human body is the best indicator as to whether or not these conditions are realized.

Temperature and relative humidity are important factors in producing comfort; the human body is to a great extent influenced by the temperature of the surrounding air, and by the rate at which perspiration is evaporated from the body into the air, which again is influenced by the relative humidity of the air.

It is generally considered that the dry-bulb temperature to produce a sense of comfort to a person at rest is 68 to 70 deg. fahr., provided a proper relation between the dry and wet-bulb temperatures is maintained.

The human organism is very susceptible to abrupt changes such as might be experienced when passing from outdoors on a cold day into a heated room in which the relative humidity is below normal or vice versa.

A ventilating system, to produce conditions of comfort and health, should therefore provide for maintaining a satisfactory relation between temperature and humidity. This relation, with a room temperature of 68 to 70 deg. fahr., generally assumes a relative humidity not below 40 per cent, nor over 60 per cent. Although this assumption is entirely traditional, a relation of humidity to temperature may be found between the limits of which true comfort will result.

Investigations from time to time by various engineering organizations and civic bodies regarding ventilating methods employed in public buildings, and particularly in schools, have disclosed the fact that systems of complete hot-blast heating and ventilation have inherent defects. Many former

advocates of this type of equipment now favor the more modern types of "split system."

It has been proved improper from the standpoint of health and comfort to employ a small quantity of highly heated air to replace the heat lost by transmission. The air supply should be large in volume and comparatively low in temperature in order to obtain the best ventilating effect. The nearer the temperature of the incoming air corresponds to the room temperature to be maintained, the more nearly is the ideal condition obtained.

To compensate for the heat losses through wall and glass and other exposures, direct radiating surface should be installed. This direct radiating surface, if placed under the windows, will also overcome the difficulties due to "outside wall and window chill" which, in the hot-blast system of heating, has been a source of considerable discomfort.

The close relation of ventilation and heating makes necessary a discussion as to the effect of various methods of ventilation upon the design of the heating plant. To illustrate these effects, some of the commonest applications of ventilation may be classified as follows:

The fireplace.
Direct-indirect system of heating and ventilation.
Indirect system of gravity ventilation.
Ventilating systems for school buildings.
Ventilating systems of large theatres and auditoriums.
Ventilation of churches.
Ventilation of banquet halls, dining rooms, kitchens, etc.
Exhaust ventilation of industrial plants.
Hot-blast systems of heating for industrial plants.

The Fireplace: The purpose of fireplaces is twofold, *first*, ornamental effect, and *second*, utility for warming at times when the heating plant is not in operation. Incidentally, also, the flue or chimney of the fireplace acts as a vent, the chimney effect or flue draft causing continuous outflow of air from the room into the atmosphere.

This outflow of air from the room through the chimney of the fireplace has the tendency of lowering the air temperature and pressure in the room, causing a greater infiltration of air from outdoors than would take place without the fireplace. The additional air finding its way into the room tends to lower the temperature, unless compensation is provided in the form of sufficient additional radiating surface.

Direct-indirect System of Heating and Ventilation: This method of heating and ventilation, as described in Chapter 6, has come into quite general use in certain sections of the country for ventilating school buildings, public libraries and courthouses.

Indirect System of Gravity Ventilation: Heating by the indirect system, in which the heat is conveyed entirely by air to the space to be heated, also provides a fair means of ventilation, but is open to the objection of highly heated incoming air.

The amount of air to be circulated is generally stipulated, which requires knowing the temperature to which the incoming air is to be heated

so that in cooling from incoming to maintained room temperature, enough heat units will be provided to offset the heat losses through windows, walls, and other exposures.

In designing heating plants of the indirect type, the total air to be circulated must be known within a fair degree of accuracy in order to determine the quantity of steam required.

The indirect method of heating requires from three to four times the quantity of steam that would be needed with direct radiation for the same warming effect. This indicates the importance of carefully considering ventilating problems in connection with heating systems, in order to determine proper proportions for boilers, pipes, radiator supply valves, return traps, and any other heating system apparatus which would be affected by the increased steam requirement due to the ventilating equipment.

With the indirect system it is also necessary to provide aspirating radiators in the vent flues.

The method of computing indirect radiating surface for given heating effects and requirements is discussed in Chapter 6.

Ventilating Systems for School Buildings: The direct-indirect and the indirect systems of heating previously mentioned are frequently used for ventilating school houses of the smaller type, but for buildings of larger proportions mechanical systems of ventilation are generally installed.

The necessity for healthful and comfortable conditions in school buildings has been the main stimulus for enacting ventilating laws by various states and cities.

Great progress has been made in late years in the design of ventilating plants for school buildings. The antiquated hot-blast system of heating and ventilation without provision for humidification has been almost entirely abandoned and superseded by the modern split-system method of ventilating with tempered air, washed and humidified before being delivered into the rooms. Direct radiation is installed for taking care of the heat lost through direct exposures of walls, windows, doors, etc.

Air is generally supplied to the class rooms through registers or diffusers placed at a level of seven to eight feet above the floor with the vent registers near the floor. The most satisfactory arrangement is generally obtained where the heat and vent flues are placed in the corridor walls and the air is blown towards the windows. The vitiated air is discharged from the vent flues into ventilators in the roof to the atmosphere.

The cold air intake should preferably be at a point above the roof. The intake openings are dampered, and additional air intake openings are provided in the attic space, making the re-circulation of air from the building possible during the heating-up period in the morning. Delivering the air into the rooms at nearly the temperature to be maintained and with automatic temperature control or modulation supply valves on the direct radiators, gives ideal conditions as near as obtainable.

In computing the requirements for direct heating in the ventilated spaces, it is only necessary to take into account the heat losses due to exposures. Exceptions, however, must be made of rooms which are to be in use after the ventilating system is shut down, such as libraries, reading rooms and offices.

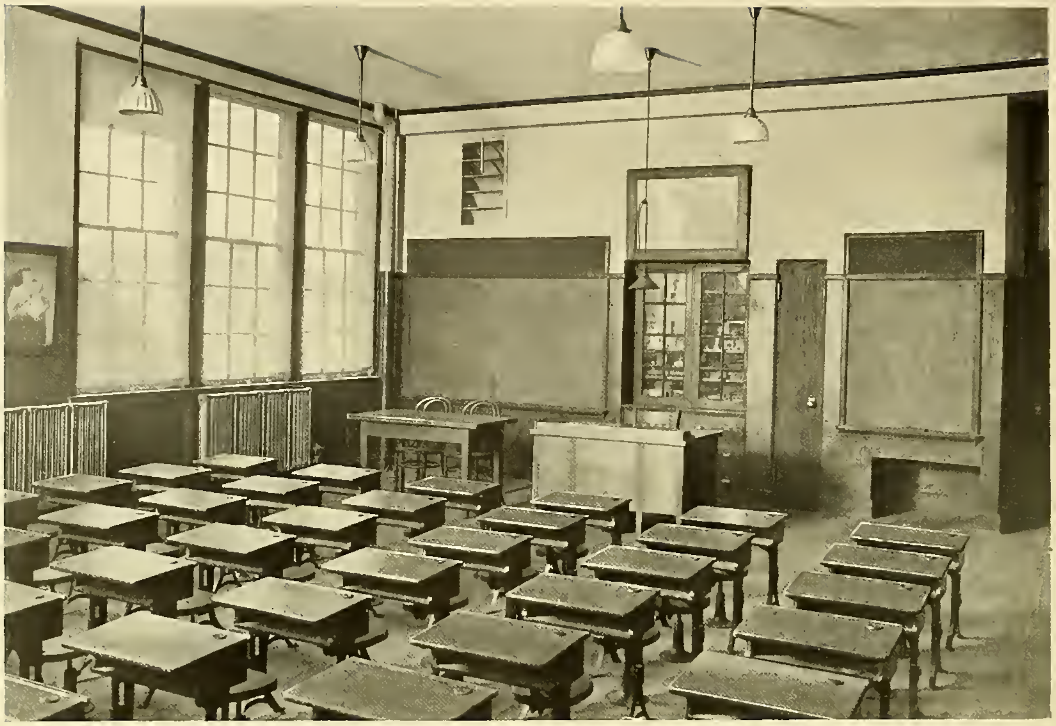


Fig. 7-1. Arrangement of fresh air inlet with diffusers, vent outlet and direct radiators in a modern school room

Ventilating systems of school buildings are usually shut down after the close of the afternoon session. Any rooms that may be in use after that period should have sufficient direct radiation to take care of the maximum requirements without the assistance of the ventilating system.

The steam required to temper the air needed for the ventilating system is generally greatly in excess of that required for the direct system of heating.

Where air washers and humidity-control systems are installed, additional steam is required to add to the heat in the air, compensating for the drop in temperature in passing through the air washer and to supply the humidity control apparatus.

Masonry ducts under floors, if used for the main trunk supply system for air distribution, should be so constructed that they can be kept dry at all times. This can be accomplished by the use of a reliable system of waterproofing. The cooling effect of these masonry ducts must be considered in the design of heating and ventilating plants and during the heating-up period sufficient time should be allowed for heating the ducts thoroughly.

The entire heating plant, including boilers, vacuum pumps, piping system and direct radiation, is affected by the method of ventilation. In the design of the plant all phases of the application and operation of the ventilating system must therefore be known and analyzed to make possible a well balanced system.

Ventilation of Theatres and Auditoriums: The ventilation of theatres and auditoriums presents an entirely different problem from that encountered in the ventilation of a building subdivided into a number of comparatively small rooms.

The problem of proper air distribution in large spaces with seating capacities numbering into thousands requires special study to provide the required quota of fresh air for each occupant.

Ventilating systems for theatres and auditoriums are usually operated only during the performances, so that portions of the structure which are in use at other times should be heated by direct radiation.

The quantity of air supplied to theatre auditoriums, on the basis of 30 cu. ft. per min. per occupant, is usually so large that sufficient heat is supplied by delivering the air into the space at a temperature a few degrees higher than that to be maintained. The temperature regulating system should be flexibile enough to automatically reduce the incoming air temperature when a large percentage of the seats are occupied, and in this way prevent excessive temperature rise in the room.

The modern theatre would not be complete without the installation of air washers, humidity-control system, and, for summer use, a refrigerating system for cooling the air.

The design of heating and ventilating systems for large auditoriums presents an interesting problem in engineering. One is so closely affected by the other that both should be worked out together so that the results obtained will harmonize.

Ventilation of Churches: Ventilation for churches is usually applied only to the main auditorium and Sunday-school room, the balance of the building being heated by direct radiation. Most churches are not continuously heated, and the warming-up period should on that account receive careful consideration by the designer. The ventilating system is generally operated during the Sunday services only.

Whether to use the up-flow system of air distribution or to discharge the air into the room through registers in the wall will greatly depend on the size of the room to be ventilated. In large churches, a combination of both, blowing in the air partly through openings in the floors in the aisles, and partly through registers in the walls, will give good results. Vent openings are usually placed in the walls near the floor and in the ceiling.

The ventilating system for a church should supply air for ventilation only and no attempt should be made to use the fan system for heating. For satisfactory results, sufficient direct radiation should be provided to compensate for all heat losses due to direct exposures and infiltration. Arrangement for re-circulating the air before the building is occupied will be found a convenience, both from the standpoint of shortening the warming-up period and also of effecting a considerable economy in the fuel consumption.

It is considered good practice to have a separate boiler and piping system for that part of the heating and ventilating plant which will be in use Sundays only, having another boiler to heat the portions of the church in use during week days.

Ventilation of Banquet Halls, Dining Rooms, Meeting Rooms, Etc.: In no other class of ventilated rooms is the efficiency or inefficiency of the ventilating system so noticeable as in banquet halls, dining rooms and meeting rooms. Smoke-laden air indicates that the ventilating system is not functioning properly, while if the air is clear and fresh in spite of smoking by the guests, a satisfactory diffusion of air in the room is shown.

As already pointed out in connection with other ventilating problems, the air should be brought in as nearly at room temperature as possible, and if heating of the room involves consideration of outside exposures, direct radiation should be used. The location and distribution of the exhaust openings is of prime importance and the exhaust should be accomplished by mechanical means. Vent openings should be placed near both floor and ceiling, and, if the structural conditions permit, additional vents should be provided in the ceiling toward the center of the room.

Kitchens require a very large air change, which should be accomplished by means of exhaust fans. Ordinances of some cities specify a three-minute air change for hotel kitchens, requiring a separate steel vent stack to be extended through the roof for this purpose. An exhaust fan, with inlet connected to this vent shaft, is usually placed in the penthouse. Above the point where the fan inlet connection is made, a tight-fitting damper propped

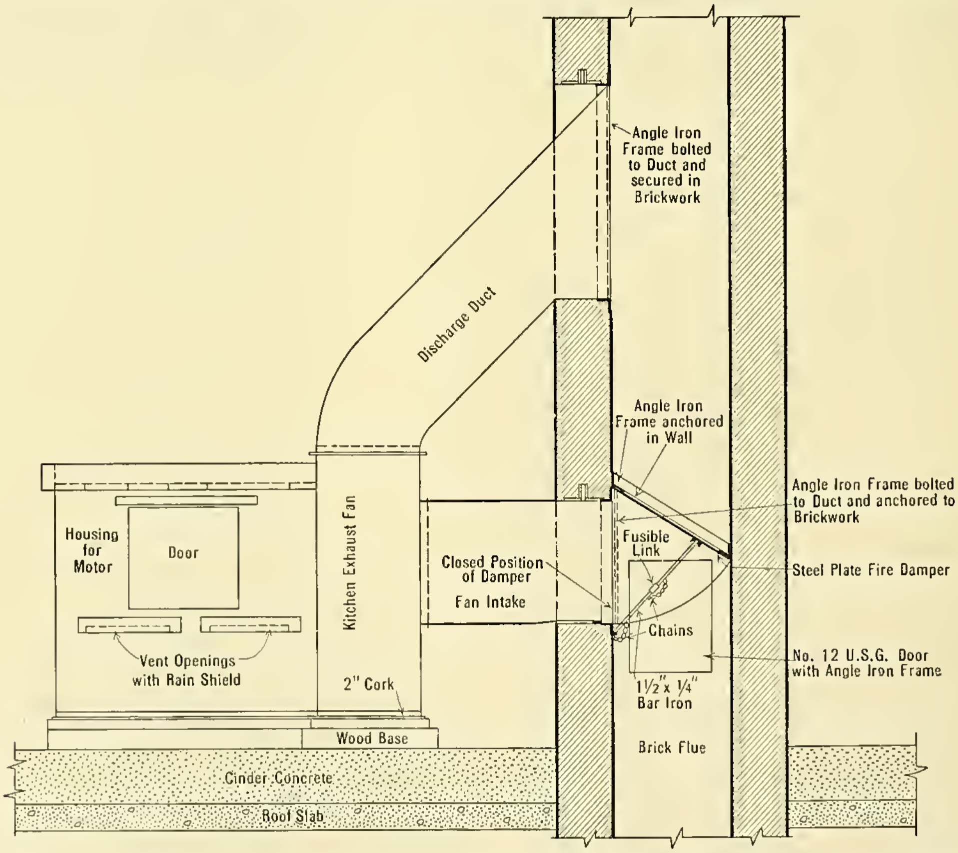


Fig. 7-2. Arrangement of fan, vent stack and safcty damper of ventilating equipment for a kitchen

open with bar iron having a fusible link is placed in the vent shaft, and the fan discharge is reconnected to the vent shaft above this damper. In case the fusible link is melted, the damper in the fan intake drops by gravity, closing the fan inlet and the stack is opened to the atmosphere. This permits the stack to burn out without damaging the exhaust fan.

Where kitchens adjoin the dining rooms, the latter can conveniently be exhausted through the kitchen. This greatly reduces the inflow of air from outdoors into the kitchen and at the same time prevents odors from the kitchen from flowing into the dining room.

Where existing conditions do not permit induction of air from warmed spaces to replace that exhausted, the air must necessarily find its way into the kitchen from outdoors and provision must be made to prevent a drop below the desired temperature. This is best accomplished by installing direct or indirect radiation for heating to the temperature needed.

Considerable heat is produced by the ranges and steam cooking utensils, so that the kitchen may be overloaded with radiation unless complete information is available as to the kitchen equipment to be used.

Exhaust Ventilation of Industrial Plants: Industries, which in their operations produce dust, acid fumes, or in any other way contaminate the air, require positive means for removing the dust or fume-laden air from the premises. Mechanical systems of exhaust ventilation are used to maintain a continuous air change by exhausting the dust-laden air.

Various types of machines, such as grinders, buffers and wood-working machines, are provided with sheet-metal ducts running to the exhaust fans, which are usually centrally located, and discharge either into dust-collecting chambers or into the atmosphere, depending upon the nature of the dust or refuse to be handled.

The continuous exhausting of air from any space will cause a corresponding inflow of outdoor air which must be heated to avoid lowering the inside temperature.

If the ventilated spaces have outside exposures, the air is drawn directly from outdoors, and infiltration takes place uniformly over the entire exposed area. A sufficient amount of direct heating surface to heat this air to the temperature to be maintained must be added to the heating surface required for heating the space without the exhaust system.

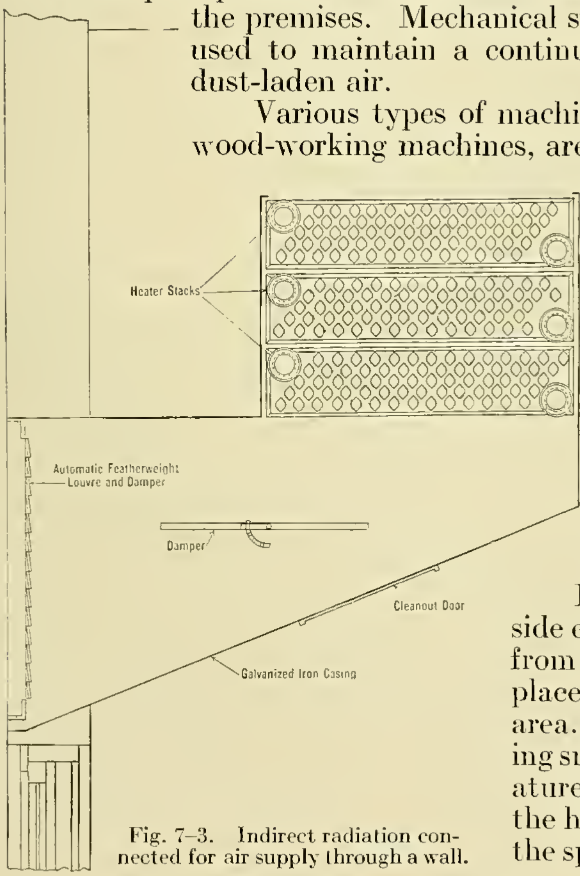


Fig. 7–3. Indirect radiation connected for air supply through a wall.

The use of large indirect radiation connected for air supply through window or other opening in outside wall, as shown in Fig. 7-3, has been found in practice to be an excellent method for warming the infiltrated air necessary to replace that removed by an exhaust fan system. In connection with temperature control of the warmed air this method has proved highly efficient.

If, however, the ventilated space has no direct exposure and connects with other rooms so that the air will be drawn from these, the additional radiation must be placed in the rooms from which the air is drawn or indirect inlets must be provided.

Chemical plants requiring the removal of acid fumes must usually exhaust large volumes of air from the rooms, and an equivalent quantity of air must be admitted directly from outdoors. This air is generally admitted through special openings in the walls and is drawn through tempering coils, so that it enters the room at the temperature to be maintained. In such cases the heating-up requirement can be eliminated from the heat loss calculations, and the direct radiation should be sufficient only to compensate for the losses through direct exposures and infiltration. However, where the exhaust system is in use only at intervals, allowances for heating up the contents of the room should be made in figuring the warming-up period. Sufficient direct radiation should be added to supply the heat units required for this purpose.

Hot-blast Systems of Heating for Industrial Plants: In industrial structures, such as large foundries, machine shops, erecting shops and round-houses, the hot-blast system of heating, instead of the direct method, is often selected, owing to its lower first cost. From the operating standpoint, however, the hot-blast system is considerably more expensive than the direct, because of the greater amount of steam required for heating by

Fig. 7–1. Arrangement of hot-air ducts of hot-blast system in an industrial plant. The side walls are protected by direct radiation placed under windows

any indirect method. This condition is particularly apparent in cases where all the air is taken directly from outdoors and after being circulated through the space is discharged into the atmosphere.

Where air can be taken from the space to be heated and re-circulated, instead of taking it from outdoors, the steam requirements are considerably reduced. In either case, the air must be heated at the fan to such a temperature that in cooling from the air-outlet temperature to that maintained inside, all heat losses are offset under maximum conditions.

Only a few general ventilating problems and their direct effect upon heating plant design have been mentioned in this chapter, but these show the importance of analyzing each problem thoroughly and making all necessary provisions for the ventilating system in heating system design.

Factors Entering Design of Complete Heating and Ventilating Plant

Air Quantities Required for Ventilation: Air quantities in many states and municipalities are fixed by legal restrictions which must be followed. However, some of the generally accepted standards are mentioned here.

The type of building and the purpose for which it is to be used are the main factors entering into the design of any ventilating system, not only as to the type of ventilation which is best adapted to each particular problem, but also as to the volume of air required.

Tables 7-1, 7-2, and 7-3 list kinds of buildings, together with their air requirements and allowable air velocities. These quantities, with slight variation, have been universally adopted.

Table 7-1. Air Requirements of Various Buildings

Type of building		Air supply. Cu. ft. per occupant per hr.
School buildings		1800
Theatre and assembly halls		1500
Churches		1500
Prisons		2100
Hospitals	Ordinary	2600
	Wounded	3500
	Contagion	6000
Residence		1600 to 2000
Factories		2000 to 3000

Table 7-2. Allowable Air Velocities. Public Building Work. Fan Systems

Supply air		Exhaust air	
Cold-air intake	700–1000 ft. per min.	Register outlets	300–400 ft. per min.
Cloth filters	About 40 " " "	Vertical flues (masonry)	400 " " "
Air washers	500 " " "	Vertical flues (sheet-metal)	500 " " "
Indirect heaters (Vento)	800–1200 " " "	Horizontal ducts	600 " " " at far end up to 1000 at fan inlet.
Horizontal air ducts	1000-1200 " " " at fan, decreasing to 600 ft. at base of flues.	Fan discharge outlet	700–1000 ft. per min.
Vertical flues (masonry)	500 ft. per min.		
Vertical flues (sheet-metal)	600 " " "		
Register outlets	200–300 " " "		

For air outlets 15 ft. or more above floor velocity may be as high as 350 ft. per min. if not thrown directly down on persons below.

Table 7-3. Allowable Air Velocities in Various Buildings in Feet per Minute

	Horizontal ducts	Vertical risers	Outlets
Factories	1500 to 2800	900 to 1500	600 to 1200
Schools	1000 to 1800	500 to 750	300 to 500
Hospitals	1000 to 1800	500 to 750	300 to 600
Theatres	1000 to 1800	500 to 750	300 to 600
Churches	1000 to 1800	500 to 750	300 to 600

Sizing of the Ducts: Two methods of estimating are in common use:

First, the velocity method, in which the velocity is fixed in the various portions of the system, and decreases from the fan outlet to the various points of discharge. This method is applicable in single-duct systems and in public buildings layouts, where the law requires certain velocity standards.

Referring to the duct design in Fig. 7-5, certain volumes and velocities are given. To determine the size of ducts at any particular point, the volume in cubic feet of air passing that point is divided by the velocity in feet at that point, which gives the required area in square feet.

Determination of the friction in any part of the duct is made by reference to the friction chart, Figure 7-7.

In a single-duct system, the longest duct, or the duct requiring greatest pressure, should be designed for certain velocities and the total pressure required at the plenum chamber determined from the friction chart, Figure 7-7. All other ducts should then be designed for the same pressure.

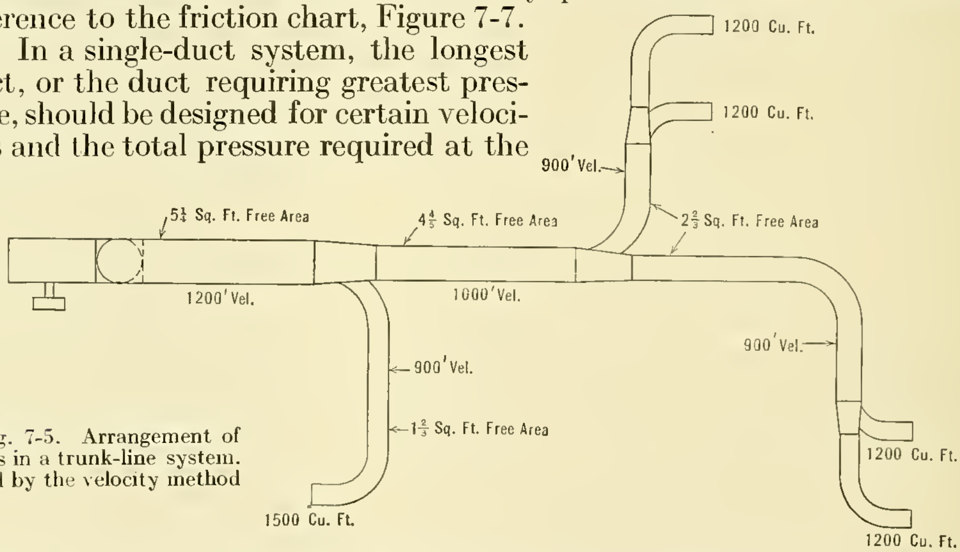


Fig. 7-5. Arrangement of ducts in a trunk-line system. Sized by the velocity method

Second—The friction-loss method, in which the duct is proportioned for equal friction pressure loss in every foot of run.

This method of duct sizing necessitates assumption of the velocity and volume at the outlets, and is adaptable to trunk-line duct systems such as are common in factories.

Table 7-4 gives an easy and accurate method for sizing ducts by pressure loss method. An example of its application follows (See Figure 7-7):

Assuming a 1000 cu. ft. discharge from each outlet at 1000 ft. velocity per min. the area of the outlet is 1 sq. ft. or say 14 in. in diameter.

Referring to Table 7-4, a 14-in. pipe is equivalent to 737 1-in. pipes

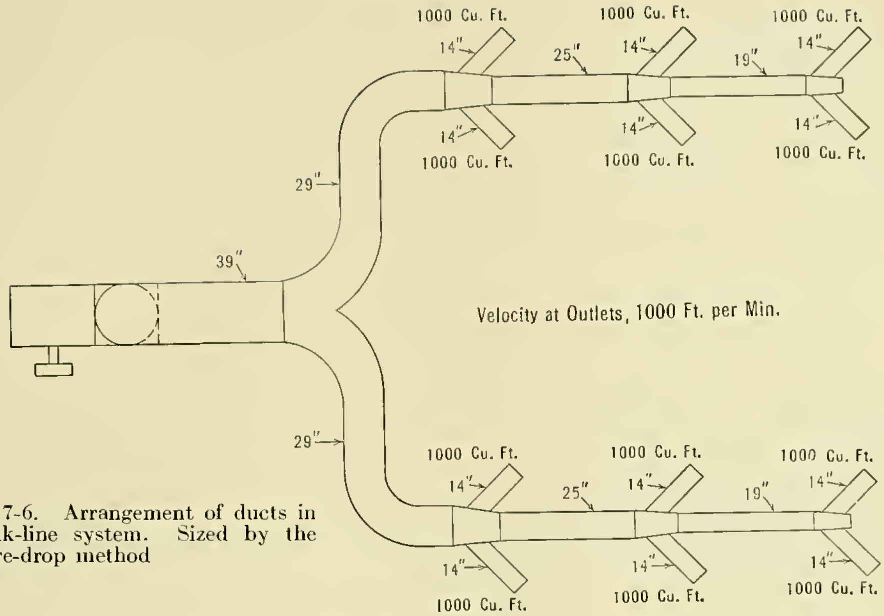


Fig. 7-6. Arrangement of ducts in a trunk-line system. Sized by the pressure-drop method

and two 14-in. pipes are equivalent to 1474 1-in. pipes. Also, 1474 1-in. pipes are equivalent to approximately a 19-in. pipe, and so on. To determine velocity at any point, the volume there is divided by the area in sq. ft. To determine friction in any portion of duct refer to Fig. 7-7.

Calculation of Resistance or Pressure: It is not the intention to go into the many complex formulae entering into the loss of pressure in ducts but rather to arrange some easily workable method.

Table 7-4. Comparison of the Air-carrying Capacity of Various Sizes of Pipes with That of a 1-in. Pipe of Same Length and Equal Friction Pressure Loss

Example—With an equal pressure loss and equal length, a 4-in. diameter pipe will carry the same volume of air as thirty-two 1-in. pipes.

Diam.	1" Pipes	Diam.	1" Pipes	Diam.	1" Pipes	Diam.	1" Pipes	Diam.	1" Pipes
1	1	21	1985	41	10565	61	28850	81	59122
2	5	22	2250	42	11300	62	30200	82	60831
3	16	23	2525	43	12030	63	31350	83	62540
4	32	24	2800	44	12621	64	32500	84	64249
5	56	25	3060	45	13400	65	33975	85	66396
6	88	26	3425	46	14100	66	35300	86	68542
7	129	27	3738	47	15000	67	36600	87	70687
8	180	28	4100	48	15850	68	38000	88	72833
9	244	29	4410	49	16610	69	39275	89	74979
10	317	30	4898	50	17600	70	40250	90	77125
11	402	31	5312	51	18275	71	41995	91	79271
12	501	32	5631	52	19335	72	43740	92	81416
13	613	33	6154	53	20000	73	45449	93	83562
14	737	34	6675	54	21500	74	47158	94	85708
15	876	35	7075	55	22300	75	48887	95	87854
16	1026	36	7735	56	23450	76	50576	96	89999
17	1197	37	8265	57	24500	77	52285		
18	1375	38	8715	58	25600	78	53995		
19	1580	39	9350	59	26700	79	55704		
20	1775	40	10060	60	27700	80	57413		

The friction chart, Figure 7-7, (based on accepted pressure loss formulae) provides quick, accurate determination of pressure loss.

Example: Assume that 30000 cu. ft. of air per minute is passed through a duct 40 in. in diameter and 50 ft. long. From the 30000 cu. ft. division at the right of chart, trace horizontally to intersection with the line representing 40 in. diameter pipe. Perpendicularly down from this point the

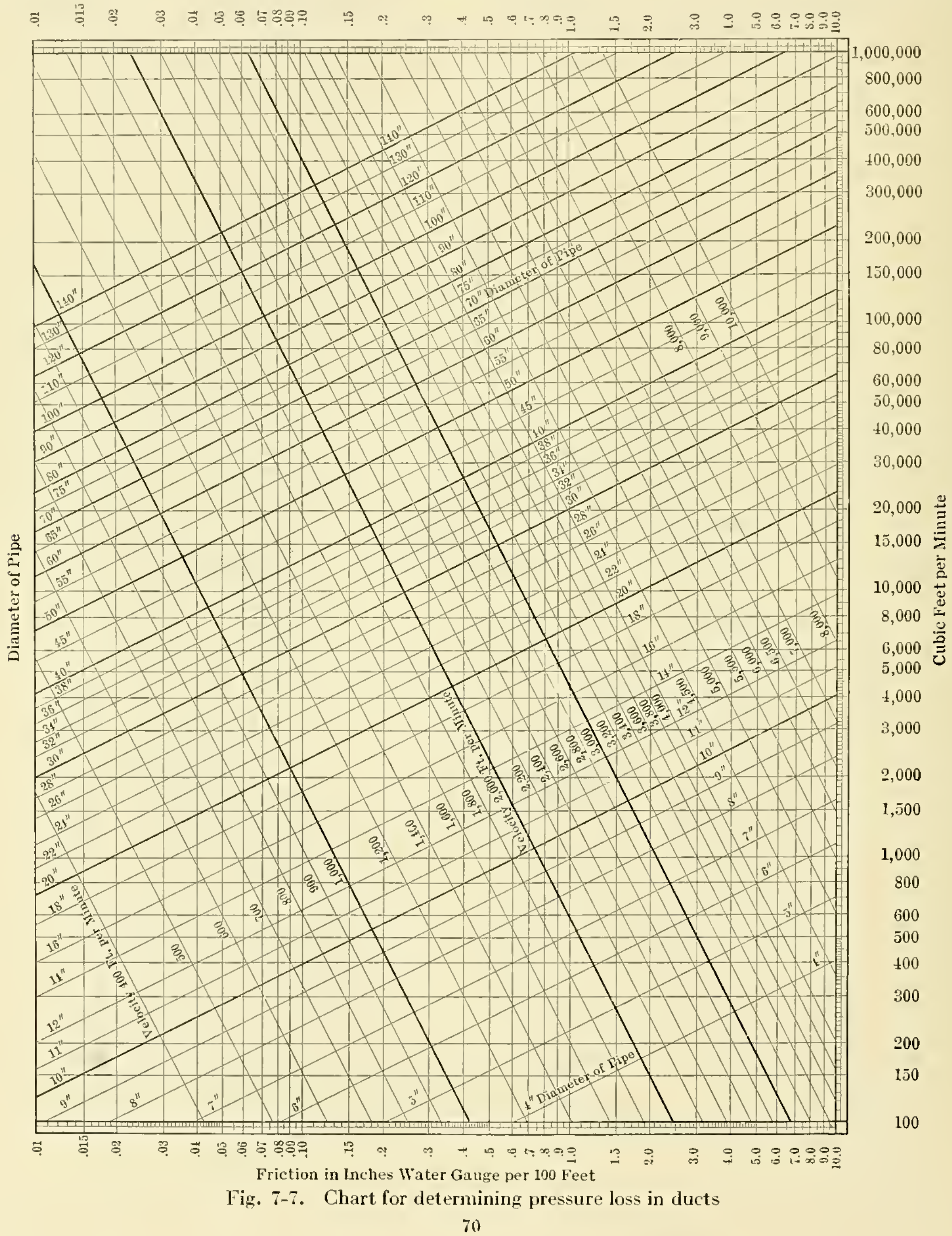


Fig. 7-7. Chart for determining pressure loss in ducts

Table 7-5. Resistance of 90-deg. Elbows

Radius of throat of elbow in diameters of pipe	Number of diameters of straight pipe offering equivalent resistance	Radius of throat of elbow in diameters of pipe	Number of diameters of straight pipe offering equivalent resistance
¼	67.0	2½	4.5
½	30.0	3	4.8
¾	16.0	3¼	5.0
1	10.0	4	5.2
1¼	7.5	4½	5.5
1½	6.0	5	5.8
1¾	5.0	5½	6.0
2	4.3		

friction in inches of water per 100 ft. of pipe is given—in this case 0.54 inches.

For 50 ft. the friction will be 50 per cent of 0.54 or 0.27 in. of water. Friction in inches of water multiplied by 0.58 gives friction in ounces.

The resistance (Table 7-5) is expressed as that of the number of diameters of straight pipe of same diameter as the elbow, and is given for elbows having different radii of throat, also expressed in diameters of pipe. For instance, a 90-deg. elbow of 24-in. pipe, having a radius of throat equal to 1 diameter, that is 24 inches, offers the same resistance to the flow of air as 10 diameters of straight pipe or 20 ft. of straight pipe.

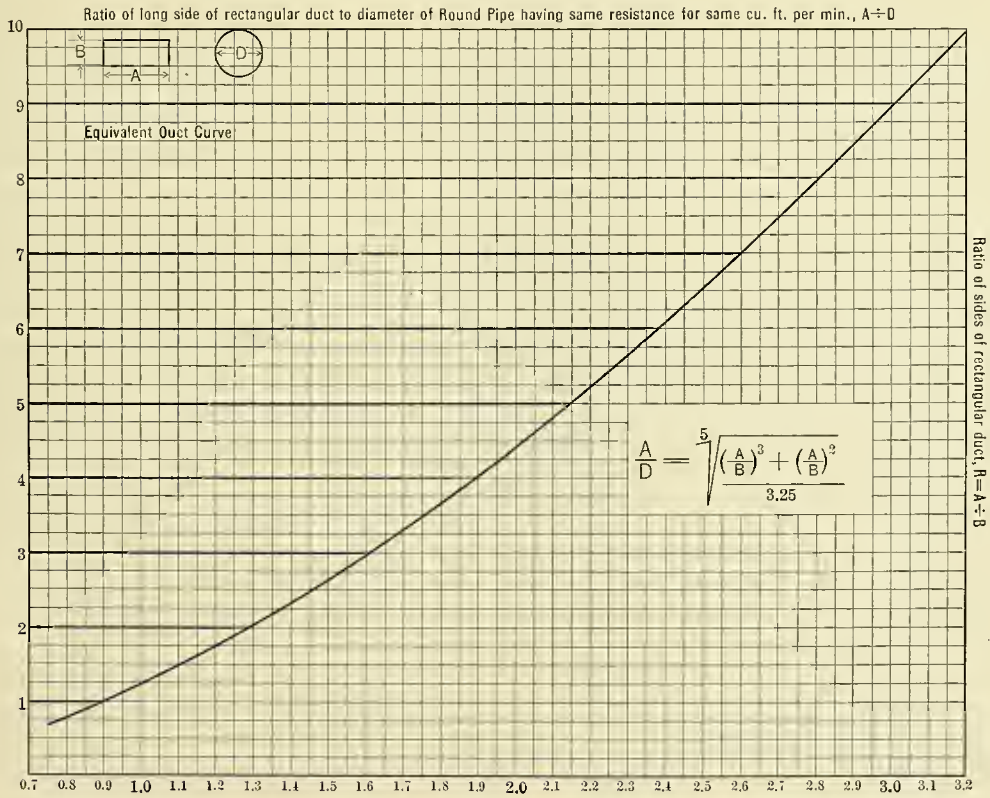


Fig. 7-8. Curve for determining the diameters of round pipes having the same friction loss for same capacity as rectangular ducts of various dimensions

To the resistance of the duct system should be added the resistance through tempering and reheating coils, also air washers, plus a small factor of safety, thereby determining the total pressure against which the fan must deliver the specified volume of air.

Where each branch duct leaves the trunk line, there should be a volume damper with trunnion, quadrant and locking device, for balancing the system.

Figure 7-8 is a curve for determining diameter of round pipe having same friction for same capacity as rectangular ducts of varying dimensions.

Selecting the Apparatus

Sizes and Arrangement of Fans: For fan performances and capacities, reference should be made to tables issued by the manufacturers.

Table 7-6. Quantities of Air at Various Temperatures Which Will Be Raised 1 deg. fahr. by 1 B.t.u.

Specific heat of air at constant pressure is 0.2375. At zero 1 cu. ft. of dry air weighs 0.0864 lb. and $\frac{1 \text{ lb.}}{.0864} = 11.574$ cu. ft. $\frac{11.574}{.2375} = 48.74$ cu. ft. raised 1 deg. by 1 B.t.u.

Temp. air deg. fahr.	Weight of 1 cu. ft.	Cu. ft. in 1 lb.	Cu. ft. 1 B.t.u. will raise 1 deg. fahr.	Temp. air deg. fahr.	Weight of 1 cu. ft.	Cu. ft. in 1 lb.	Cu. ft. 1 B.t.u. will raise 1 deg. fahr.	Temp. air deg. fahr.	Weight of 1 cu. ft.	Cu. ft. in 1 lb.	Cu. ft. 1 B.t.u. will raise 1 deg. fahr.
0	.0864	11.58	48.71	72	.0747	13.39	56.40	152	.0649	15.40	64.90
12	.0842	11.87	50.00	82	.0733	13.64	57.40	162	.0638	15.65	66.00
22	.0824	12.14	51.00	92	.0720	13.90	58.60	172	.0628	15.90	67.00
32	.0807	12.40	52.20	102	.0707	14.14	59.20	182	.0618	16.17	68.00
42	.0791	12.64	53.10	112	.0694	14.40	60.60	192	.0609	16.42	69.10
52	.0776	12.88	54.10	122	.0682	14.65	61.60	202	.0600	16.67	70.10
62	.0761	13.13	55.20	132	.0671	14.90	62.80	212	.0591	16.92	71.30
70	.0750	13.34	56.30	142	.0660	15.15	63.80				

Heaters: To select a heater for any set of conditions it is necessary to know the volume of air to be handled, its initial temperature, and the temperature to which it must be raised.

Two methods for determining the above quantities are available where the building is heated as well as ventilated by the air. One applies where a definite air change is desired or where ventilation must be provided for a given number of people.

Example: Assume a building requiring 18000 cu. ft. per min. measured at 70 deg. fahr. with a total of 860000 B. t. u. loss through exposed glass, walls, etc. Then

$$\frac{\text{B.t.u. loss per hr.}}{\text{Cu. ft. per min.} \times .2375 \times .075 \times 60} = \text{diffusion}$$

$$= \frac{860{,}000}{18000 \times .2375 \times .075 \times 60} = 45 \text{ deg. fahr.}$$

45 deg. diffusion + 10 deg. duct loss + 70 deg. desired room temperature = 125 deg. final temperature at coils. In this calculation 0.2375 is the specific heat of air and is constant and 0.075 is the weight of one cubic foot of air at the room temperature of 70 deg. (See Table 7-6.)

The other method is to decide on the final temperature to be used with some fixed entering temperature.

Example: Suppose the hourly heat loss through exposed walls, glass, etc., is 1204500 B.t.u. Assume a final temperature at the heater of 135 deg. fahr. and a loss of 10 deg. in the ducts. The temperature at the duct outlets will then be 125 deg. fahr. The room temperature desired is 65 deg. and the outside temperature is 0 deg.

The difference in the temperature between the duct outlets and the room temperature is available for heating.

$$\text{Cu. ft. per min.} = \frac{\text{B.t.u. per hr.}}{60 \times 60 \times .2375 \times .068} = \frac{1204500}{60 \times 60 \times .2375 \times .068} =$$

20720 cu. ft. per min. required, in which 0.2375 is specific heat of air and is constant and 0.068 is weight of one cu. ft. of air at 125 deg. (See Table 7-6.)

Either of the above formulæ can be used on split systems where a portion of the losses through walls, glass, etc., are taken care of by direct radiation, and the balance by the incoming air. In the split system where all heat loss through walls, glass, etc., is taken care of by direct radiation, the final temperature of the air is, of course, the same as the room temperature desired. However, in choosing the heater, allowance should be made for some temperature drop in the ducts (usually 10 to 20 degrees).

After determining the volume and final temperature of the air the size of heater can readily be chosen from tables furnished by manufacturers.

Table 7-7. B.t.u. Required for Heating Air*

This table specifies the quantity of heat in B. t. u. required to raise 1 cu. ft. of air through any given temperature interval

Temperature of air in room, deg. fahr.

External Temp.	40°	50°	60°	70°	80°	90°	100°	110°	120°	130°
−40°	1.802	2.027	2.252	2.479	2.703	2.928	3.154	3.379	3.604	3.829
−30°	1.540	1.760	1.980	2.200	2.420	2.640	2.860	3.080	3.300	3.520
−20°	1.290	1.505	1.720	1.935	2.150	2.365	2.580	2.795	3.010	3.225
−10°	1.051	1.262	1.473	1.684	1.892	2.102	2.311	2.522	2.732	2.943
0°	0.822	1.028	1.234	1.439	1.645	1.851	2.056	2.262	2.467	2.673
10°	0.604	0.805	1.007	1.208	1.409	1.611	1.812	2.013	2.215	2.416
20°	0.393	0.590	0.787	0.984	1.181	1.378	1.575	1.771	1.968	2.165
30°	0.192	0.385	0.578	0.770	0.963	1.155	1.345	1.540	1.733	1.925
40°	0.000	0.188	0.376	0.564	0.752	0.940	1.128	1.316	1.504	1.692
50°	0.000	0.000	0.184	0.367	0.551	0.735	0.918	1.102	1.286	1.470
60°	0.000	0.000	0.000	0.179	0.359	0.538	0.718	0.897	1.077	1.256
70°	0.000	0.000	0.000	0.000	0.175	0.350	0.525	0.700	0.875	1.049

* F. Schumann's *Manual af Heating and Ventilation.*

Boiler Horsepower Required: To determine the boiler horsepower required for air heating, the following formula can be used:

$$\frac{\text{Cu. ft. per min.} \times 60 \times A}{B} = \text{lb. steam per hour.}$$

in which A = B.t.u. required for heating 1 cu. ft. of air from initial to final temperature (See Table 7-7).

B = latent heat of steam

$$\frac{\text{lb. steam per hr.}}{34.5} = \text{boiler horsepower}$$

From the manufacturers' tables the condensation rates per square foot of surface are given for various velocities and temperatures, and it is well to check up the above formula from these given factors.

CHAPTER VIII

Proportioning of Chimneys

NO problem in the heating of buildings presents greater elements of uncertainty than that of properly proportioning the chimney.

In larger installations, such as isolated plants for the production of power, light and heat, the conditions may usually be very accurately determined in advance. By use of the formula given hereafter, proper results follow in almost every case.

A. Chimneys for House-Heating Boilers

In small plants and particularly residence heating, it is not practicable to make such accurate advance determinations of all the conditions. Usually the chimney is built into the wall, thereby requiring that its cross-section must be proportioned to the width of brick. Chimneys so built are usually either smoothly mortared on the inside or lined with thin tile of rectangular or circular cross-section. The latter gives such freedom from friction and eddy currents and lessened surface for loss of heat in the gases, that a round chimney lining will frequently give fully as good results as would be obtained in the square of brick-work in which it is enclosed.

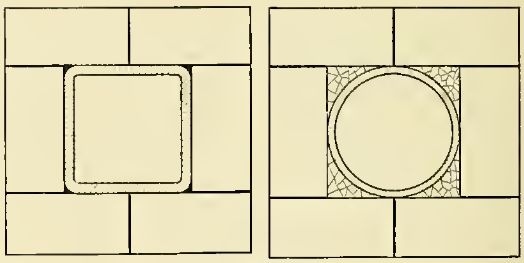


Fig. 8-1. Cross-sections through typical house chimneys

The inclination to cut down cross-sectional area to save cost and space in the portion of building through which the chimney passes should be discouraged as false economy. Once the chimney is built into the structure, increase of area is practically impossible, and a chimney that is too small

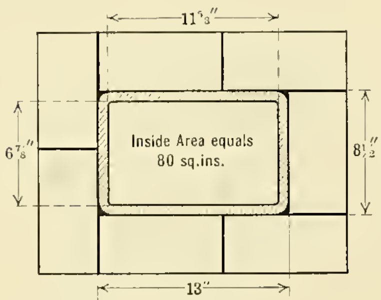


Fig. 8-2. Seven bricks per course

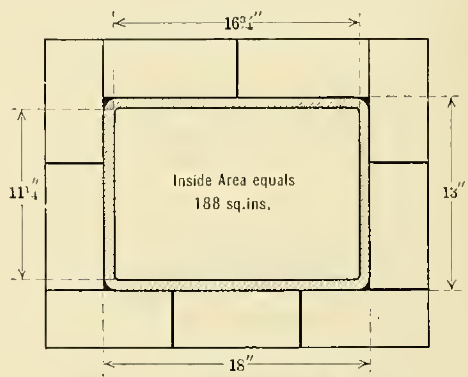


Fig. 8-3. Nine bricks per course

remains a source of discomfort and waste during the entire life of the structure. Little is saved in building an 8½-in. by 13-in. flue as compared with a 13-in. by 18-in. flue, the latter having more than twice the area and more than twice the capacity, while the bricks per course are as 9 is to 7. (See Figures 8-2 and 8-3.)

To get the greatest effectiveness, a definite amount of draft must be available. The actual amount required varies widely for different types of commercial cast-iron boilers, and, unfortunately, it is not always possible to know in advance which make of these boilers will be selected or may later be installed. It is, therefore, preferable to provide for excessive draft which may be controlled by damper, rather than to risk insufficient draft, the remedying of which is almost hopeless.

For ascertaining the probable interior cross-section of round or rectangular flue linings, also unlined brick chimneys necessary for average cast-iron heating boilers where height in feet from combustion chamber to top of chimney and maximum hourly rate of evaporation in pounds of water are known, Figures 8-7a and 8-7b will be found convenient.

With the maximum rate and height of chimney determined, enter the table at right-hand column at the determined hourly evaporation rate; fol-

Table 8-1. Dimensions of Flue Linings

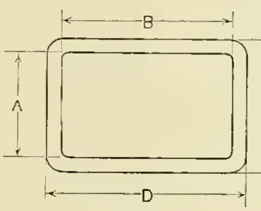


Fig. 8-4

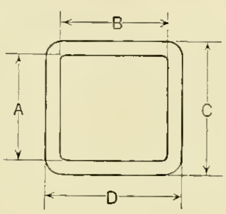


Fig. 8-5

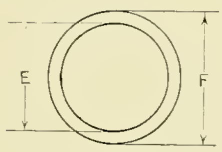


Fig. 8-6

As manufactured by

The Delaware Clay Products Co. Pittsburgh, Penna.								W. S. Dickey Clay Mfg. Co. Kansas City, Mo.								Robinson Clay Products Co. Akron, Ohio							
Rectangular and square					Circular			Rectangular and square					Circular			Rectangular and square					Circular		
Sq. in. free area	A	B	C	D	Sq. in. free area	E	F	Sq. in. free area	A	B	C	D	Sq. in. free area	E	F	Sq. in. free area	A	B	C	D	Sq. in. free area	E	F
23	3¼	7¼	4½	8½	28	6	7¼	29	3¾	7¾	4½	8½				23	3¼	7	4¾	8⅝	28	6	7⅛
36	3⅛	11⅝	4½	13	38	7	8⅜	61	7 13/16	7 13/16	8½	8½				36	3 1/16	11¾	4¾	13¼	38	7	8½
47	2⅞	16⅜	4½	18	50	8	9½	46	3¾	12¼	4½	13				60	3⅞	15½	4½	17	50	8	9
39	6¼	6¼	7½	7½	64	9	10 5/16	92	7⅝	12⅛	8½	13				47	4½	10½	6	12	64	9	10½
52	7 3/16	7 3/16	8½	8½	78	10	11¾	145	12 1/16	12 1/16	13	13				33	5¾	5¾	7¼	7¼	78	10	12
80	6 15/16	11 7/16	8½	13	113	12	14	127	7⅝	16⅝	8½	17½	125	12⅝	14¼	52.5	7¼	7¼	8½	8½	113	12	14
110	6¾	16¼	8½	18	176	15	17¼	202	12⅛	16⅝	13	17½				80	6⅞	11⅝	8½	13	176	15	17⅛
129	11⅜	11⅜	13	13	254	18	20½	270	16 7/16	16 7/16	17½	17½				104	6½	16	8½	18	254	18	20⅞
188	11¼	16¾	13	18	314	20	22¾						291	19¼	21½	127	11¼	11¼	13	13	314	20	23
256	16	16	18	18	380	22	25¼									169	10¾	15¾	13	18	346	21	
					452	24	27¼						499	25 3/16	27½	240	15½	15½	18	18	380	22	
																					452	24	27
																					572	27	
																					707	30	35
																					855	33	
																					1018	36	

Note. All dimensions are in inches and subject to slight variation

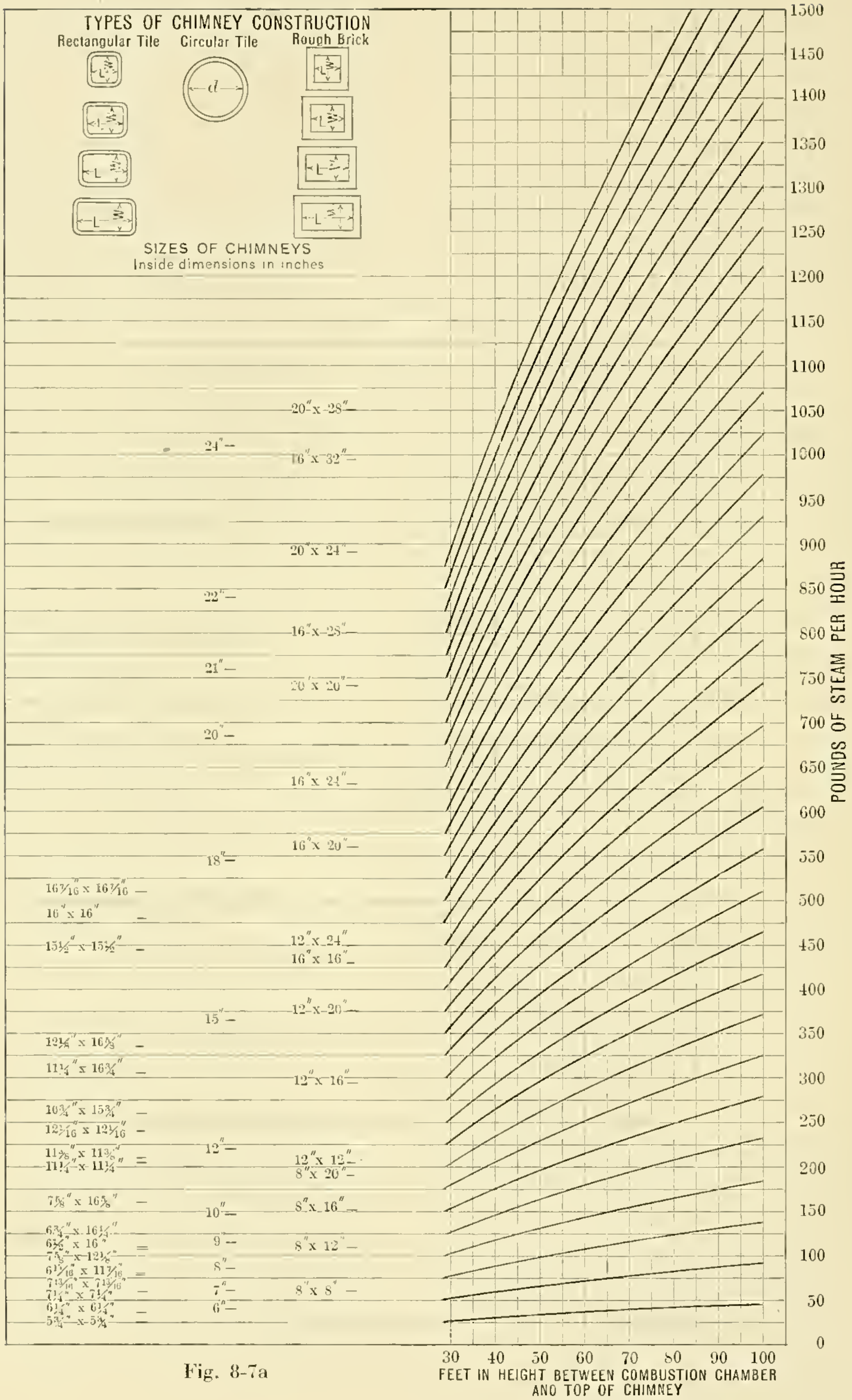

TYPES OF CHIMNEY CONSTRUCTION
Rectangular Tile
Circular Tile
Rough Brick
SIZES OF CHIMNEYS
Inside dimensions in inches
POUNDS OF STEAM PER HOUR
FEET IN HEIGHT BETWEEN COMBUSTION CHAMBER AND TOP OF CHIMNEY

Fig. 8-7a

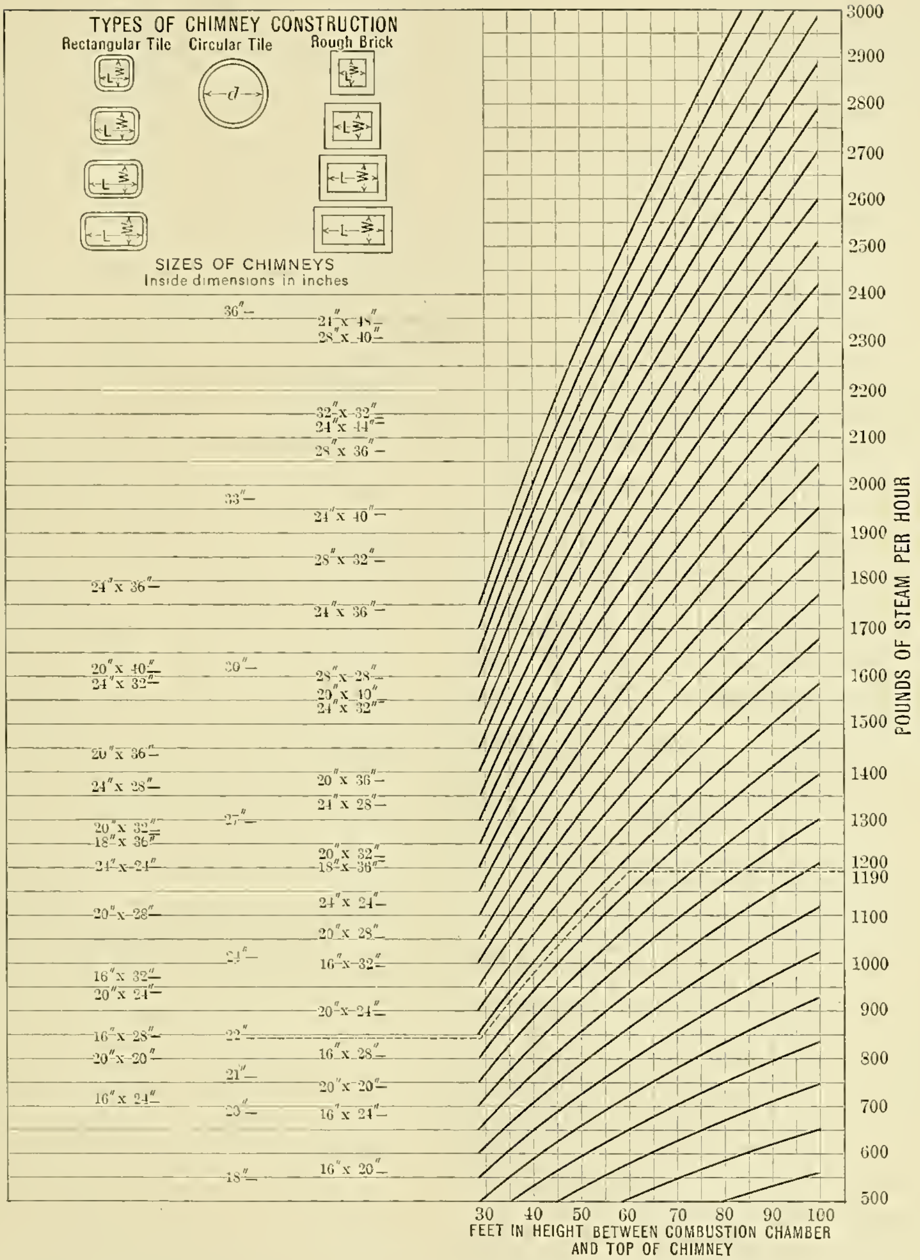


Fig. 8-7b—Probable capacities of chimneys of different forms, sizes and heights to produce proper draft for average cast-iron boiler of up-draft type using anthracite coal

low horizontally to left to intersection of vertical line representing given height, then downward along the curve to its left end, then follow a horizontal line to left; the interior cross-sections of linings and rough brick above the horizontal should be ample under usual conditions.

When desiring to ascertain probable capacity of a chimney of known dimensions and construction, the chart is read in reverse order.

Dotted lines on Fig. 8-7b indicate that for 11800 lb. evaporation per hr. and 60-ft. chimney height, a 22-in. diameter, or 16 by 28-in. tile lining should be proper, or that 20 by 24-in. rough brick would be ample.

It must, however, be borne in mind that the location of the building in relation to topography and surrounding structures may render a chimney absolutely inefficient, while another similar in every respect of height and cross-section, used for similar boiler and fuel, but favorably located, will be able to produce a superabundance of draft; also that the resistance due to thickness of coal bed, character and quality of fuel as well as resistance between the combustion chamber and chimney, vary in different makes of boilers having similar ratings, and that these resistances form a large part of the total head for which chimneys are required.

The chimney problem should be presented to the boiler manufacturer for his study and recommendation.

B. Chimneys and Draft for Power Boilers*

The height and diameter of a properly designed chimney depend upon the amount of fuel to be burned, the design of the flue, with its arrangement relative to the boiler or boilers, and the altitude of the plant above sea level. There are so many factors involved that as yet there has been produced no formula which is satisfactory in taking them all into consideration and the methods used for determining stack sizes are largely empirical. In this chapter a method sufficiently comprehensive and accurate to cover all practical cases will be developed and illustrated.

Draft is the difference in pressure available for producing a flow of the gases. If the gases within a stack be heated, each cubic foot will expand, and the weight of the expanded gas per cubic foot will be less than that of a cubic foot of the cold air outside the chimney. Therefore, the unit pressure at the stack base due to the weight of the column of heated gas will be less than that due to a column of cold air. This difference in pressure, like the difference in head of water, will cause a flow of the gases into the base of the stack. In its passage to the stack the cold air must pass through the furnace or furnaces of the boilers connected to it, and it in turn becomes heated. This newly heated gas also rises in the stack and the action is continuous.

The intensity of the draft, or difference in pressure, is usually measured in inches of water. Assuming an atmospheric temperature of 62 deg. fahr. and the temperature of the gases in the chimney as 500 deg. fahr., and, neglecting for the moment the difference in density between the chimney gases and the air, the difference between the weights of the external air and the internal flue gases per cubic foot is 0.0347 lb., obtained as follows:

Weight of a cubic foot of air at 62 deg. fahr. = 0.0761 lb.
Weight of a cubic foot of air at 500 deg. fahr. = 0.0414 lb.

Difference = 0.0347 lb.

* Reprinted from *Steam* by permission of Babcock & Wilcox Co.

Therefore, a chimney 100 ft. high, assumed for the purpose of illustration to be suspended in the air, would have a pressure exerted on each square foot of its cross-sectional area at its base of 0.0347 x 100 = 3.47 lb. As a cubic foot of water at 62 deg. fahr. weighs 62.32 lb., an inch of water would exert a pressure of 62.32 ÷ 12 = 5.193 lb. per sq. ft. The 100-ft. stack would, therefore, under the above temperature conditions, show a draft of 3.47 ÷ 5.193 or approximately 0.67 in. of water.

The method best suited for determining the proper proportion of stacks and flues is dependent upon the principle that if the cross-sectional area of the stack is sufficiently large for the volume of gases to be handled, the intensity of the draft will depend directly upon the height; therefore, the method of procedure is as follows:

(1) Select a stack of height to produce the draft required by the particular character of fuel and amount burned per square foot of grate surface.

(2) Determine the cross-sectional area necessary to handle the gases without undue frictional losses.

The application of these rules follows:

Draft Formula: The force or intensity of the draft, not allowing for difference in density of air and of the flue gases, is given by the formula:

$$D = 0.52\ H \times P\left(\frac{1}{T} - \frac{1}{T_1}\right) \qquad \textit{(Formula 8-1)}$$

in which

D = draft produced, measured in inches of water,
H = height of top of stack above grate bars in feet,
P = atmospheric pressure in pounds per square inch,
T = absolute atmospheric temperature,
T_1 = absolute temperature of stack gases.

In this formula no account is taken of the density of the flue gases, it being assumed that it is the same as that of air. Any error arising from this assumption is negligible in practice, as a factor of correction is applied in using the formula to cover the difference between the theoretical figures and those corresponding to actual operating conditions.

The force of draft at sea level (which corresponds to an atmospheric pressure of 14.7 lb. per sq. in.) produced by a chimney 100 ft. high with the temperature of the air at 60 deg. fahr. and that of the flue gases at 500 deg. fahr. is,

$$D = 0.52 \times 100 \times 14.7\left(\frac{1}{521} - \frac{1}{961}\right) = 0.67$$

Under the same temperature conditions this chimney at an atmospheric pressure of 10 lb. per sq. in. (which corresponds to an altitude of about 10000 ft. above sea level) would produce a draft of,

$$D = 0.52 \times 100 \times 10\left(\frac{1}{521} - \frac{1}{961}\right) = 0.45$$

For using this formula it is handy to tabulate values of the product,

$$0.52 \times 14.7\left(\frac{1}{T} - \frac{1}{T_1}\right)$$

which we will call K, for various values of T_1. With these values calculated for assumed atmospheric temperature and pressure, Formula 8-1 becomes,

$$D = K\ H. \qquad (Formula\ 8\text{-}2)$$

For average conditions the atmospheric pressure may be considered 14.7 lb. per sq. in., and the temperature 60 deg. fahr. For these values and various stack temperatures K becomes:

Temperature of stack gases	*Constant K*
750	.0084
700	.0081
650	.0078
600	.0075
550	.0071
500	.0067
450	.0063
400	.0058
350	.0053

Draft Losses: The intensity of the draft as determined by the above formula is theoretical and can never be observed with a draft gauge or any recording device. However, if the ashpit doors of the boiler are closed and there is no perceptible leakage of air through the boiler setting or flue, the draft measured at the stack base will be approximately the same as the theoretical draft. The difference existing at other times represents the pressure necessary to force the gases through the stack against their own inertia and the friction against the sides. This difference will increase with the velocity of the gases. With the ashpit doors closed the volume of gases passing to the stack is a minimum and the maximum force of draft will be shown by a gauge.

As draft measurements are taken along the path of the gases, the readings grow less as the points at which they are taken are farther from the stack, until in the boiler ashpit, with the ashpit doors open for freely admitting the air, there is little or no perceptible rise in the water of the gauge. The breeching, the boiler damper, the baffles and the tubes, and the coal on the grates all retard the passage of the gases, and the draft from the chimney is required to overcome the resistance offered by the various factors. The draft at the rear of the boiler setting where connection is made to the stack or flue may be 0.5-in., while in the furnace directly over the fire it may not be over, say, 0.15-in., the difference being the draft required to overcome the resistance offered in forcing the gases through the tubes and around the baffling.

One of the most important factors to be considered in designing a stack is the pressure required to force the air for combustion through the bed of fuel on the grates. This pressure will vary with the nature of the fuel used, and in many instances will be a large percentage of the total draft. In the case of natural draft, its measure is found directly by noting the draft in the furnace, for with properly designed ashpit doors it is evident that the pressure under the grates will not differ sensibly from atmospheric pressure.

Loss in Stack: The difference between the theoretical draft as determined by Formula 8-1 and the amount lost by friction in the stack proper, is the available draft, or that which the draft gauge indicates when

connected to the base of the stack. The sum of the losses of draft in the flue, boiler and furnace must be equivalent to the available draft, and as these quantities can be determined from record of experiments, the problem of designing a stack becomes one of proportioning it to produce a certain available draft.

The loss in the stack due to friction of the gases can be calculated from the following formula:

$$\triangle D = \frac{fW^2CH}{A^3} \qquad \textit{(Formula 8-3)}$$

in which

$\triangle D$ = draft loss in inches of water,
W = weight of gas in pounds passing per second,
C = perimeter of stack in feet,
H = height of stack in feet,
f = a constant with the following values at sea level:
.0015 for steel stacks, temperature of gases 600 deg. fahr.
.0011 for steel stacks, temperature of gases 350 deg. fahr.
.0020 for brick or brick-lined stacks, temperature of gases 600 deg. fahr.
.0015 for brick or brick-lined stacks, temperature of gases 350 deg. fahr.
A = area of stack in square feet.

This formula can also be used for calculating the frictional losses for flues, in which case, C = the perimeter of the flue in feet, H = the length of the flue in feet, the other values being the same as for stacks.

The available draft is equal to the difference between the theoretical draft from Formula 8-2 and the loss from Formula 8-3, hence:

$$d^1 = \text{available draft} = KH - \frac{fW^2CH}{A^3} \qquad \textit{(Formula 8-4)}$$

Table 8-2 gives the available draft in inches that a stack 100 ft. high will produce when serving different horsepowers of boilers with the methods of calculation for other heights.

Height and Diameter of Stacks: From Formula 8-4, it becomes evident that a stack of certain diameter, if it be increased in height, will produce the same available draft as one of larger diameter, the additional height being required to overcome the added frictional loss. It follows that among the various stacks that would meet the requirements of a particular case there must be one which can be constructed more cheaply than the others. It has been determined from relation of stack costs to diameters and heights, in connection with the formula for available draft, that the minimum cost stack has a diameter dependent solely upon the horsepower of the boilers served, and a height proportional to available draft required.

Assuming 120 lb. of flue gas per hr. for each boiler horsepower, which provides for ordinary overloads and use of poor coal, the method stated gives:

For unlined steel stack—diameter in inches $= 4.68\,(\text{hp.})^{\frac{2}{5}}$. *(Formula 8-5.)*
For masonry lined stack—diameter in inches $= 4.92\,(\text{hp.})^{\frac{2}{5}}$. *(Formula 8-6.)*

In both of these formulae, hp. = the rated horsepower of the boiler.

From this formula the curve, Figure 8-8, has been calculated and from it the stack diameter for any boiler horsepower can be selected.

Table 8-2. Available Draft

Calculated for 100-ft. stack of different diameters, assuming stack temperature of 500 deg. fahr. and 100 lb. of gas per hp. For other heights of stack multiply draft by height ÷ 100

Horse-power	Diameter of stack in inches														
	36	42	48	54	60	66	72	78	84	90	96	102	108	114	120
100	.64														
200	.55	.62													
300	.41	.55	.61												
400	.21	.46	.56	.61											
500		.34	.50	.57	.61										
600		.19	.42	.53	.59										
700			.34	.48	.56	.60	.63								
800			.23	.43	.52	.58	.61	.63							
900				.36	.49	.56	.60	.62	.64						
1000				.29	.45	.53	.58	.61	.63	.64					
1100					.40	.50	.56	.60	.62	.63	.64				
1200					.35	.47	.54	.58	.61	.63	.64	.65			
1300					.29	.44	.52	.57	.60	.62	.63	.64	.65		
1400						.40	.49	.55	.59	.61	.63	.64	.65	.65	
1500						.36	.47	.53	.58	.60	.62	.63	.64	.65	.65
1600						.31	.43	.52	.56	.59	.62	.63	.64	.65	.65
1700							.41	.50	.55	.58	.61	.62	.64	.64	.65
1800							.37	.47	.54	.57	.60	.62	.63	.64	.65
1900							.34	.45	.52	.56	.59	.61	.63	.64	.64
2000								.43	.50	.55	.59	.61	.62	.63	.64
2100								.40	.49	.54	.58	.60	.62	.63	.64
2200								.38	.47	.53	.57	.59	.61	.62	.64
2300								.35	.45	.52	.56	.59	.61	.62	.63
2400								.32	.43	.50	.55	.58	.60	.62	.63
2500									.41	.49	.54	.57	.60	.61	.63

Horse power	Diameter of stack in inches							
	90	96	102	108	114	120	132	144
2600	.47	.53	.56	.59	.61	.62	.64	.65
2700	.45	.52	.55	.58	.60	.62	.64	.65
2800	.44	.50	.55	.58	.60	.61	.64	.65
2900	.42	.49	.54	.57	.59	.61	.63	.65
3000	.40	.48	.53	.56	.59	.61	.63	.64
3100	.38	.47	.52	.56	.58	.60	.63	.64
3200		.45	.51	.55	.58	.60	.63	.64
3300		.44	.50	.54	.57	.59	.62	.64
3400		.42	.49	.53	.56	.59	.62	.64
3500		.40	.48	.52	.56	.58	.62	.64
3600			.47	.52	.55	.58	.61	.63
3700			.45	.51	.55	.57	.61	.63
3800			.44	.50	.54	.57	.61	.63
3900			.43	.49	.53	.56	.60	.63
4000			.42	.48	.52	.56	.60	.62
4100			.40	.47	.52	.55	.60	.62
4200			.39	.46	.51	.55	.59	.62
4300				.45	.50	.54	.59	.62
4400				.44	.49	.53	.59	.62
4500				.43	.49	.53	.58	.61
4600				.42	.48	.52	.58	.61
4700				.41	.47	.51	.57	.61
4800				.40	.46	.51	.57	.60
4900					.45	.50	.57	.60
5000					.44	.49	.56	.60

For other stack temperature add or deduct before multiplying by $\frac{\text{height}}{100}$ as follows:*

For 750 deg. fahr. add .17 in.	For 650 deg. fahr. add .11 in.	For 550 deg. fahr. add .04 in.	For 400 deg. fahr. deduct .09 in.
For 700 deg. fahr. add .14 in.	For 600 deg. fahr. add .08 in.	For 450 deg. fahr. deduct .04 in.	For 350 deg. fahr. deduct .14 in.

* Results secured by this method will be approximately correct

For stoker practice where a large stack serves a number of boilers, the area is usually made about one-third more than the above rules call for, which allows for leakage of air through the setting of any idle boilers, irregularities in operating conditions, etc.

Stacks with diameters determined as above will give an available draft which bears a constant ratio of the theoretical draft, and allowing for the cooling of the gases in their passage upward through the stack, this ratio is 0.8. Using this factor in Formula 8-2, and transposing, the height of the chimney becomes,

$$H = \frac{d^1}{.8K} \qquad \textit{(Formula 8-7)}$$

Where H = height of stack in feet above the level of the grates,
d^1 = available draft required,
K = constant as in Formula 8-2.

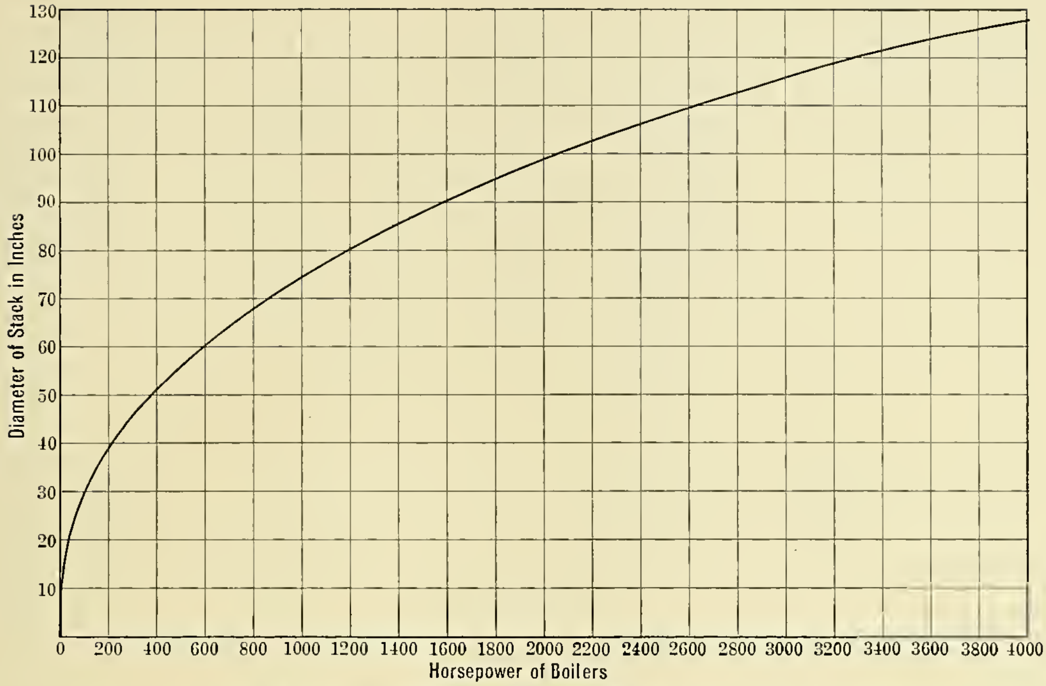


Fig. 8-8. Diameter of stacks and horsepower they will serve
Computed from Formula (8-5). For brick or brick-lined stacks increase the diameter 6 per cent

Losses in Flues: The loss of draft in straight flues due to friction and inertia can be calculated approximately from Formula 8-3, which was given for loss in stacks. It is to be borne in mind that C in this formula is the actual perimeter of the flue and is least, relative to the cross-sectional area, when the section is a circle, is greater for a square section, and greatest for a rectangular section. The retarding effect of a square flue is 12 per cent greater than that of a circular flue of the same area and that of a rectangular with sides as 1 and 1½, 15 per cent greater. The greater resistance of the more or less uneven brick or concrete flue is provided for in the value of the constants given for Formula 8-3. Both steel and brick flues should be short and should have as near a circular or square cross-section as possible. Abrupt turns are to be avoided, but as long easy sweeps require valuable space, it is often desirable to increase the height of the stack rather than to take up added space in the boiler room. Short right-angle turns reduce the draft by an amount which can be roughly approximated as equal to 0.05-in. for each turn. The turns which the gases make in leaving the damper box of a boiler, in entering a horizontal flue and in turning up into a stack should always be considered. The cross-sectional areas of the passages leading from the boilers to the stack should be of ample size to provide against undue frictional loss. It is poor economy to restrict the size of the flue and thus make additional stack height necessary to overcome the added friction. The general practice is to make flue areas the same or slightly larger than that of the stack; these should be, preferably, at least 20 per cent greater, and a safe rule to follow in figuring flue areas is to allow 35 sq. ft. per 1000

horsepower. It is unnecessary to maintain the same size of flue the entire distance behind a row of boilers, and the areas at any point may be made proportional to the volume of gases that will pass that point. That is, the areas may be reduced as connections to various boilers are passed.

With circular steel flues of approximately the same size as the stacks, or reduced proportionally to the volume of gases they will handle, a convenient rule is to allow 0.1-in. draft loss per 100 ft. of flue length and 0.05-in. for each right-angle turn. These figures are also good for square or rectangular steel flues with areas sufficient to provide against excessive frictional loss. For losses in brick or concrete flues, these figures should be doubled.

Underground flues are less desirable than overhead or rear flues for the reason that in most instances the gases will have to make more turns where underground flues are used and because the cross-sectional area of such flues will oftentimes be decreased on account of an accumulation of dirt or water which it may be impossible to remove.

In tall buildings, such as office buildings, it is frequently necessary in order to carry spent gases above the roofs to install a stack the height of which is out of all proportion to the requirements of the boilers. In such cases it is permissible to decrease the diameter of a stack, but care must be taken that this decrease is not sufficient to cause a frictional loss in the stack as great as the added draft intensity due to the increase in height, which local conditions make necessary.

In such cases also the fact that the stack diameter is permissibly decreased is no reason why flue sizes connecting to the stack should be decreased. These should still be figured in proportion to the area of the stack that would be furnished under ordinary conditions or with an allowance of 35 sq. ft. per 1000 horsepower, even though the cross-sectional area appears out of proportion to the stack area.

Loss in Boilers: In calculating the available draft of a chimney, 120 lb. per hr. has been used as the weight of the gases per boiler horsepower. This covers an overload of the boiler to an extent of 50 per cent and provides for the use of poor coal. The loss in draft through a boiler proper will depend upon its type and baffling and will increase with the per cent of rating at which it is run. No figures can be given which will cover all conditions, but for approximate use in figuring the available draft necessary it may be assumed that the loss through a boiler will be 0.25-in. where the boiler is run at rating, 0.40-in. where it is run at 150 per cent of its rated capacity, and 0.70-in. where it is run at 200 per cent of its rated capacity.

Loss in Furnace: The draft loss in the furnace or through the fuel bed varies between wide limits. The air necessary for combustion must pass through the interstices of the coal on the grate. Where these are large, as in the case with broken coal, but little pressure is required to force the air through the bed; but if they are small, as with bituminous slack or small sizes of anthracite, a much greater pressure is needed. If the draft is insufficient the coal will accumulate on the grates and a dead, smoky fire will result with the accompanying poor combustion; if the draft is too great, the coal may be rapidly consumed on certain portions of the grate, leaving

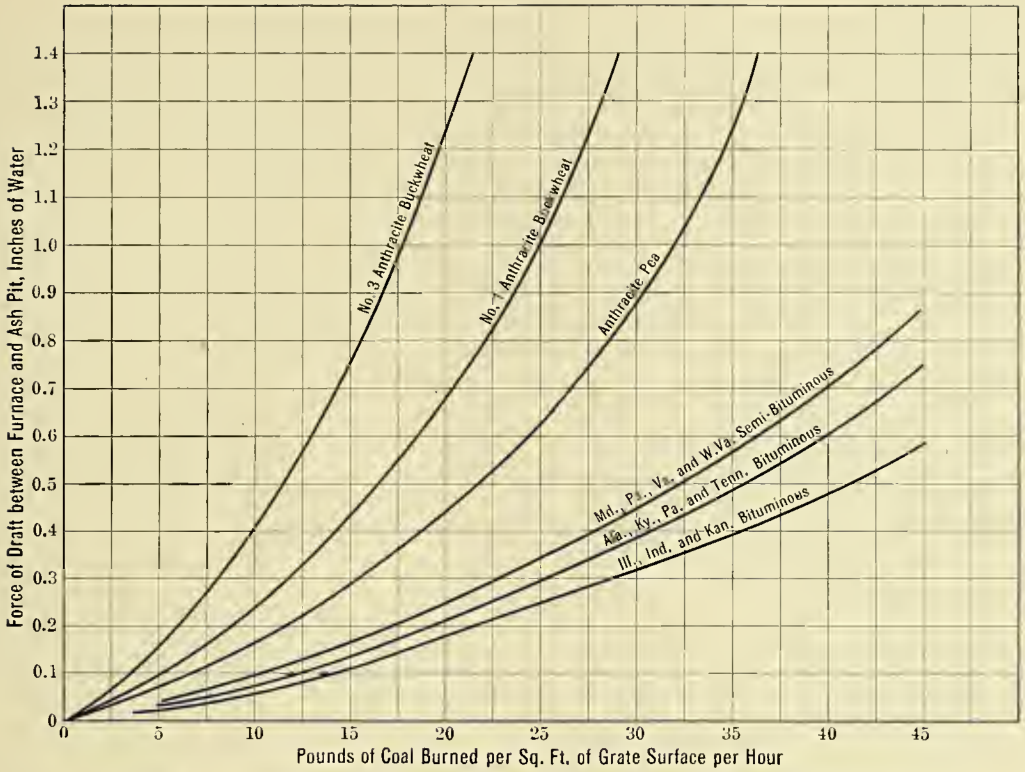


Fig. 8-9. Draft required at different combustion rates for various kinds of coal

the fire thin in spots and a portion of the grates uncovered with the resulting losses due to an excessive amount of air.

Draft Required for Different Fuels: For every kind of fuel and rate of combustion there is a certain draft with which the best general results are obtained. A comparatively light draft is best with the free-burning bituminous coals and the amount to use increases as the percentage of volatile matter diminishes and the fixed carbon increases, being highest for the small sizes of anthracites. Numerous other factors, such as the thickness of fires, the percentage of ash and the air spaces in the grates bear directly on this question of the draft best suited to a given combustion rate. The effect of these factors can only be found by experiment. It is almost impossible to show by one set of curves the furnace draft required at various rates of combustion for all of the different conditions of fuel, etc., that may be met. The curves in Figure 8-9, however, give the furnace draft necessary to burn various kinds of coal at the combustion rates indicated by the abscissae, for a general set of conditions. These curves have been plotted from the records of numerous tests and allow a safe margin for economically burning coals of the kinds noted.

Rate of Combustion: The amount of coal which can be burned per hour per square foot of grate surface is governed by the character of the coal and the draft available. Where the boiler and grate are properly propor-

tioned, the efficiency will be practically the same, within reasonable limits, for different rates of combustion. The area of the grate, and the ratio of this area to the boiler heating surface will depend upon the nature of the fuel to be burned, and the stack should be so designed as to give a draft sufficient to burn the maximum amount of fuel per square foot of grate surface corresponding to the maximum evaporative requirements of the boiler.

Solution of a Problem: The stack diameter can be determined from the curve, Figure 8-8. The height can be determined by adding the draft losses in the furnace, through the boiler and flues, and computing from Formula 8-7 the height necessary to give this draft.

Example: Proportion a stack for boilers rated at 2000 horsepower, equipped with stokers, and burning bituminous coal that will evaporate 8 lb. of water from and at 212 deg. fahr. per lb. of fuel; the ratio of boiler heating surface to grate surface being 50: 1; the flues being 100 ft. long and containing two right-angle turns; the stack to be able to handle overloads of 50 per cent; and the rated horsepower of the boilers based on 10 sq. ft. of heating surface per horsepower.

The atmospheric temperature may be assumed as 60 deg. fahr. and the flue temperatures at the maximum overload as 550 deg. fahr. The grate surface equals 400 sq. ft. The total coal burned at rating $= \frac{2000 \times 34\frac{1}{2}}{8}$ $= 8624$ lb. The coal per square foot of grate surface per hour at rating $= \frac{8624}{400} = 22$ lb.

For 50 per cent overload the combustion rate will be approximately 60 per cent greater than this, or $1.60 \times 22 = 35$ lb. per sq. ft. of grate surface per hr. The furnace draft required for the combustion rate, from the curve, Figure 8-9, is 0.6-in. The loss in the boiler will be 0.4-in., in the flue 0.1 in., and in the turns $2 \times 0.05 = 0.1$-in. The available draft required at the base of the stack is, therefore,

	Inches
Boiler	0.4
Furnace	0.6
Flues	0.1
Turns	0.1
Total	1.2

Since the available draft is 80 per cent of the theoretical draft, this draft due to the height required is $1.2 \div 0.8 = 1.5$ inches.

The chimney constant for temperatures of 60 deg. fahr. and 550 deg. fahr. is 0.0071 and from Formula 8-7,

$$H = \frac{1.5}{.0071} = 211 \text{ ft.}$$

Its diameter from curve in Figure 8-7 is 96 in. if unlined, and 102 in. inside if lined with masonry. The cross-sectional area of the flue should be approximately 70 sq. ft. at the point where the total amount of gas is to be handled, tapering to the boiler farthest from the stack to a size which will depend upon the size of the boiler units used.

Correction in Stack Sizes for Altitudes: It has been assumed that a stack height for altitude will be increased inversely as the ratio of barometric pressure at the altitude to that at sea level, and that the stack diameter increases inversely as the two-fifths power of this ratio. This relation assumes a constant draft measured in inches of water at base of stack for a given rate of boiler operation, regardless of altitude.

If the assumption be made that boilers, flues and furnaces remain the same, and further that the increased velocity of a given weight of air passing through the furnace at a higher altitude would have no effect on the combustion, the theory has been advanced* that a different law applies.

Under the above assumptions, whenever a stack is working at its maximum capacity at any altitude, the entire draft is utilized in overcoming the various resistances, each of which is proportional to the square of the velocity of the gases. Since boiler areas are fixed, all velocities may be related to a common velocity, say that within the stack, and all resistances may, therefore, be expressed as proportional to the square of the chimney velocity. The total resistance to flow, in terms of velocity head, may be expressed in terms of weight of a column of external air, the numerical value of such head being independent of the barometric pressure. Likewise the draft of a stack, expressed in height of column of external air, will be numerically independent of the barometric pressure. It is evident, therefore, that if a given boiler plant, with its stack operated with a fixed fuel, be transplanted from sea level to an altitude, assuming the temperatures remain constant, the total draft head measured in height of column of external air will be numerically constant. The velocity of chimney gases will, therefore, remain the same at altitude as at sea level and the weight of gases flowing per second with a fixed velocity will be proportional to the atmospheric density or inversely proportional to the normal barometric pressure.

To develop a given horsepower requires a constant weight of chimney gas and air for combustion. Hence, as altitude is increased, the density is decreased and, for the assumptions given, the velocity through furnace, boiler passes, breeching and flues must be correspondingly greater at altitude than at sea level. The mean velocity, therefore, for given boiler horsepower and constant weight of gases will be inversely proportional to the barometric pressure and the velocity head measured in column of external air will be inversely proportional to the square of the barometric pressure.

For stacks operating at altitude it is necessary not only to increase the height but also the diameter, as there is an added resistance within the stack due to the added friction from the additional height. This frictional loss can be compensated by a suitable increase in the diameter and when so compensated, the chimney height would have to be increased at a ratio inversely proportional to the square of the normal barometric pressure.

In designing a boiler for high altitudes, as already stated, the assumption is usually made that a given grade of fuel will require the same draft measured in inches of water at the boiler damper as at sea level, and this leads to making the stack height inversely as the barometric pressures, instead of inversely as the square of the barometric pressures. The correct height, no doubt,

* *Chimneys for Crude Oil*, C. R. Weymouth, Trans. Am. Soc. M. E., Dec., 1912

falls somewhere between the two values as larger flues are usually used at the higher altitudes, whereas to obtain the ratio of the squares, the flues must be the same size in each case, and again the effect of an increased velocity of a given weight of air through the fire at a high altitude, on the combustion, must be neglected. In making capacity tests with coal fuel, no difference has been noted in the rates of combustion for a given draft suction measured by a water column at high and low altitudes, and this would make it appear that the correct height to use is more nearly that obtained by the inverse ratio of the barometric readings than by the inverse ratio of the squares of the barometric readings. If the assumption is made that the value falls midway between the two formulæ, the error in using a stack figured in the ordinary way by making the height inversely proportional to the barometric readings, would differ about 10 per cent in capacity at an altitude of 10000 ft., which difference is well within the probable variation of the size determined by different methods. It would, therefore, appear that ample accuracy is obtained in all cases by simply making the height inversely proportional to the barometric readings and increasing the diameter so that the stacks used at high altitudes have the same frictional resistance as those used at low altitudes, although, if desired, the stack may be made somewhat higher at high altitudes than called for in order to be safe.

The increase of stack diameter necessary to maintain the same friction loss is inversely as the two-fifths power of the barometric pressure.

Table 8-3. Stack Capacities, Correction Factors for Altitudes

Altitude, height in feet above sea level	Normal barometer	R, ratio barometer reading sea level to altitude	R^2	$R^{\frac{2}{5}}$, ratio increase in stack diameter
0	30.00	1.000	1.000	1.000
1000	28.88	1.039	1.079	1.015
2000	27.80	1.079	1.164	1.030
3000	26.76	1.121	1.257	1.047
4000	25.76	1.165	1.356	1.063
5000	24.79	1.210	1.464	1.079
6000	23.87	1.257	1.580	1.096
7000	22.97	1.306	1.706	1.113
8000	22.11	1.357	1.841	1.130
9000	21.28	1.410	1.988	1.147
10000	20.49	1.464	2.144	1.165

Table 8-3 gives the ratio of barometric readings of various altitudes to sea level, values for the square of this ratio and values of the two-fifths power of this ratio. These figures show that the altitude affects the height to a much greater extent than the diameter, and that practically no increase in diameter is necessary for altitudes up to 3000 ft.

For high altitudes the increase in stack height necessary is, in some cases, such as to make the proportion of height to diameter impracticable. The method to be recommended in overcoming, at least partially, the great increase in height necessary at high altitudes is an increase in the grate surface of the boilers which the stack serves, in this way reducing the combustion rate necessary to develop a given power and hence the draft required for such combustion rate.

CHAPTER IX

Boilers

THE boiler equipment is the production center of the heating system and the point where the bulk of the operating expense is centered. For this reason, a heating plant can be successful and economical only if the boiler equipment is of correct type, good material and workmanship, well proportioned from the standpoint of its work and ample in capacity.

Service from a heating system cannot properly be termed satisfactory unless the desired heating effect is secured without waste of fuel and without excess labor at the boilers, so it is the endeavor of this chapter to promote a better understanding of the boiler parts and what they should do.

Due consideration should be given to the proper selection of a boiler, not only as to size and capacity, but also as to its adaptability to the existing local conditions which, if not properly considered, may affect the success of the entire plant.

It is not intended in this discussion to cover any details of boiler construction, which properly come under the province of, and can best be solved by, the boiler makers themselves.

Steam boilers have been built in one form or another for nearly 200 years, yet today they are the least understood of all the important elements which make up a power or heating plant.

If it were not necessary to consider the efficiency of the performance of a steam boiler, such as the number of pounds of water evaporated by a pound of fuel, or the relation of grate surface to heating surface, etc., the problem would be simple.

All the years of experience and the thousands of evaporating tests made have not produced any definite and reliable rule or formula for calculating either the amount of steam that will be generated per hour with a given fuel or the quantity of steam in pounds produced per pound of fuel burned in the furnace.

Lucke* says: "There is no absolute measure of boiler performance as to capacity or efficiency as a basis of comparison to measure the goodness of a boiler as a boiler; comparison must, therefore, be between one and another boiler, or one and another service condition; one boiler may be said to be better than another, or one condition more favorable and another worse, for the result desired, but hardly more than this is possible."

For commercial purposes, boiler capacities seem to be quite well standardized, boilers used for heating work being rated in capacity of square feet of steam radiation, and boilers for power work in boiler horse-power.

The boiler capacity rating in square feet is based on equivalent cast-iron direct radiation with condensation rate of ¼ lb. steam per sq. ft. per hr.

The American Society of Mechanical Engineers in 1885 adopted a double definition of the Boiler Horsepower as follows:

(a) The evaporation of 34.5 lb. water per hr. from and at 212 deg. fahr.

* *Engineering Thermodynamics*

(b) The absorption by water, between fuel conditions and that of the steam leaving the boiler, of 33,305 B.t.u. per hr.

A steam boiler consists of the following essential parts: A furnace in which the combustion of the fuel takes place; a vessel to contain water to be evaporated; a steam space where the steam is liberated and where the generated steam is contained; a heating surface to transmit the heat of the furnace to the water; a smoke pipe to carry away the products of combustion, and various attachments, such as gauges, damper regulators, safety valves, etc.

A proper relation of the first four parts to each other constitutes a successful heating boiler.

It is of prime importance that the furnace is of proper design as regards grate area, size of combustion chamber, ash pit, etc., to give most efficient operation, permitting the consumption of the maximum effective quantity of fuel per square foot of grate area. Further references will be made to importance of selecting the proper kind of grates for the various grades of fuel available in various localities.

The water space or the water-holding capacity of a boiler does not always receive enough attention. It should be remembered that the boiler which holds the greatest quantity of water at or near the normal water line for given size or capacity is the safest one to use, because in such a boiler the water line is not so readily brought down to and below the danger point, as compared with another having only about half the water-holding capacity.

An investigation of the various cast-iron boilers to which our remarks regarding the water-holding capacity particularly refer, will show that there is an astonishing difference in this particular feature. Selecting two boilers of the same capacity but of different makes, it will be found that the water-holding capacity at or near normal water line varies as much as 1 to 4. It stands to reason that the boiler from which 4 gal. of water can be withdrawn by lowering the water line ½ in. will be safer than the boiler which shows ½ in. lower water with loss of only 1 gallon.

Boiler manufacturers recognize more and more that if a boiler is to be successful the steam space should be liberal. The velocity with which the steam bubbles are separated from the water in the liberating space is extremely high. A boiler with limited steam-liberating surface will very likely lose its water under heavy load conditions because under the influence of this velocity, particles are carried over with the steam into the piping system.

The heating surface of a boiler includes all parts of the boiler shell, flues, tubes, etc., covered by water and exposed to hot gases. Surface having steam on one side and hot gases on the other is superheating surface.

The American Society of Mechanical Engineers recommends that in measuring heating surface, the side next to the gases be used. Thus when estimating the heating surface of water-tube boilers, the *outside* areas of the tubes are measured, and for return-tubular or fire-box boilers the *inside* areas are measured.

The heat generated by the combustion of fuel permeates from the furnace through the heating surface to the water in the boiler. As the process of combustion proceeds, the heat liberated is immediately absorbed, partly by heat from the freshly added fuel, but mainly from the gaseous products of

combustion. The absorption of heat by these substances causes a rise in their temperature and from these gases the heat is transmitted through the heating surfaces into the boiler water. This transmission of heat takes place in three distinct ways, each of which is governed by a definite law not applicable to the others.

Before the heat reaches the body of the boiler water, it changes its mode of travel at least twice. It is first imparted to the heating surface: (a) by *radiation* from the hot fuel bed, the furnace walls and the luminous flames, and (b) by convection from the hot moving gaseous products of combustion. Upon reaching the heating surfaces the heat changes its mode of transmission and passes through the soot, metal and scale to the inner surface, which is in contact with the water, purely by conduction. From the wet side of the heating surface the heat is carried into the boiler water mainly by convection.*

The water in the boiler can absorb only that heat called the "heat available for the boiler," which is above its own temperature. Heat below this temperature will not flow into the boiler and is, therefore, not available.

A commercial boiler absorbs only part of the available heat, which expressed as a percentage, is the true boiler effciency. This efficiency depends chiefly on the arrangement of the heating surfaces. Therefore, from point of economy in operation, the heating surface available and its arrangement should be carefully considered by the designer when selecting boiler equipment for a heating plant.

The boiler efficiency, which is the only true measure of the ability of the boiler to absorb heat, is expressed by the following equation:

$$\text{True boiler efficiency} = \frac{\text{heat absorbed by boiler}}{\text{heat available for boiler}}$$

The efficiencies ordinarily used in commercial boiler tests may not represent the true performance of the boiler under actual working conditions.

Boiler capacities as given in catalogues of manufacturers of heating boilers are based on the efficiencies obtained in the testing laboratories, and these may not be representative of true conditions. In selecting a boiler for a heating plant, due allowance should be made to take care of this discrepancy by adding a factor of safety to compensate for the difference in laboratory and actual working conditions. This allowance, which may be called the safety factor to be added to the theoretical capacity, varies widely for the various types of boilers. Before determining the safety factor to be added to the commercial rating, the designer should carefully consider the type of boiler, the kind of fuel to be used, and the kind of attention the plant will receive, as all these bear on the performance and efficiency.

The necessity of providing an extra safety factor is recognized also by the heating trade and various trade associations that have established rules and regulations for guidance of members in determining boiler capacities.

The difficulty in obtaining the more desirable grades of coal has resulted in an increasing tendency to use coals which are more readily obtainable and lower in cost. The grates of the boilers should therefore be

* *Bulletin 18*, United States Bureau of Mines

properly designed for the fuel which will most likely be used. Different authorities have a wide range of opinion as to the width of the air space that should be used between grate bar openings for a given grade of fuel.

Professor Gebhardt recommends an air space of ⅝ in. between the grate bars and bars ¾ in. wide for power boilers and for average bituminous coal.

For No. 3 buckwheat coal an air space of 3/16 in. and for No. 1 buckwheat 5/16 in. is recommended.

Grate areas are usually determined in proportion to the heating surface of the boiler, that is, for a given fuel, the grate surface and heating surface have a fixed ratio. For normal operation, a ratio of grate surface to heating surface of 1 to 35 to 45 develops the rated capacity of the boiler, while for fine coal or overload conditions, a ratio of 1 to 25 is desirable.

For return-tubular boilers and water-tube boilers, the following table shows the usual ratios of grate surface to heating surface and also the grate bar openings applying with these ratios when using soft coal fuel.

Table 9-1. Grate Surfaces for Soft Coals

Coal	Grate bar openings		Ratio of grate surface to heating surface.	
	Mine run	Slack	Mine run	Slack
Va., W. Va., Md., Pa.	½-in.	⅜-in.	1:55	1:50
Ohio, Ky., Tenn., Ala.	⅜ – ½	¼	1:50	1:45
Ill., Ind., Kan., Okla.	⅜ – ½	¼	1:45	1:40
Col. and Wyo.	⅜	¼	1:45	1:40

Determination of the amount of grate surface to be used under given conditions involves the available draft as well as the fuel to be used. The curves given in Figure 8-9, page 85, show how much draft is necessary for burning different coals at various rates of combustion.

The draft required to overcome resistances in the boiler is also given in Chapter 8, pages 83 and 84. These losses in the boiler and furnace must be deducted from the total available draft to determine the draft available for the fuel bed.

The capacity of the boiler and the B.t.u. to be developed being known, the number of pounds of coal to be burned can be readily computed. The total grate area required is found by dividing the total number of pounds of coal to be burned by the rate of combustion taken from Figure 8-9, page 85.

Hand-fired return tubular and water-tube boilers are readily operated at the rates of combustion in pounds of coal per square foot of grate area given in Table 9-2.

Small boilers of the residence-heating type usually burn coal at rates ranging from 1 to 5 lb. per sq. ft. of grate surface per hr. and in larger heating

Table 9-2. Rates of Combustion for Various Coals

Anthracite	15 lb. per sq. ft. per hr.
Semi-anthracite	16 " " " " " "
Semi-bituminous	18 " " " " " "
Eastern bituminous	20 " " " " " "
Western bituminous	28 " " " " " "

boilers the ratio ranges from 4 to 12 lb. per sq. ft. of grate surface per hr.

These low rates of combustion are the result of demands for less frequent attention, in order that the man who fires the boiler may devote time to other work. In consequence, heating boilers are expected to do their work when fired once every hour or two or in residence heating, once in six to eight hours, whereas power boilers are fired at regular intervals of five to ten minutes.*

Another reason why heating boilers require different firing methods to burn bituminous coals successfully is that the space in the fire-box above the fuel bed is usually very much smaller than is the corresponding space in power boilers.* This space, known as the combustion chamber, is where the smoky gases driven off from the coal must become mixed with air and burn. The more rapidly the combustible gases are driven off from the coal, the larger must be the space necessary for burning them completely. The relatively small combustion space in heating boilers makes it important that the firing be done in a way to prevent the gases from being driven off too rapidly.†

The type of boiler to fit the given conditions most satisfactorily depends upon the physical conditions of the plant, as well as the type of heating system selected. The success of one depends upon the other. For this reason boiler selection is discussed also in Chapter 10, Selection of the Proper Type of Steam Heating System.

On account of the great variation of governing conditions, no attempt will be made here to discuss in detail the method of installation of the boilers or their connections.

Precautions should be taken in the design of the boiler plant to minimize bad effects from priming.

Liberal bleeder or drip connections from the boiler header, connecting directly to the return header, eliminate a great percentage of this trouble.

Priming in most cases is due to the presence of grease or oil in the boiler or to the presence in the water of certain alkalies which cause the water to foam or bubble, and be carried into the piping system by the steam. Before it can be expected to perform its functions uniformly, effectively and economically, a boiler must be thoroughly cleansed of oil, scale, dirt and other impurities. The priming of boilers is not confined to any particular type or make. The plant designer will safeguard the interest of the owner and himself as well, if he makes sure that bleeder connections are made to protect the boiler in case of priming and that his instructions about proper cleaning of the boiler and the entire heating system are carried out in full by the heating contractor.

For thoroughly cleaning a boiler, the safety valve should be removed and a sufficient quantity of soda ash should be placed within the boiler to cause saponification of oils and grease. A temporary overflow pipe should be run to waste from the safety valve outlet or highest point of the boiler.

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* *Technical Paper 180*, United States Bureau of Mines

† For further reference to the importance and effect of combustion space see *Technical Papers 63, 80 and 137* of the United States Bureau of Mines

With a moderate fire and the addition of feed water as required to prevent injury, the foaming of the boiler will cause the flow of oil and grease through the overflow pipe to waste. After thorough boiling, the fire should be drawn and when cool, the water should be withdrawn and then the boiler should be thoroughly washed with clean water to remove dirt and chemicals. This treatment for boilers should be repeated whenever necessary as indicated by abnormal fluctuations of the water line or by the appearance of foaming.

Damper control is an important feature of boiler operation. There are two classes of damper regulators, (1) those that move the damper for slight changes in the steam pressure, with a proportional movement due to the change in pressure and (2) those that operate the dampers between extreme positions when the steam pressure changes. The first is preferable from the standpoint of economical combustion.

As mentioned in Chapter 8, the fuel in a steam-boiler furnace is made to burn by passing through it a current of air, which supplies the necessary oxygen and carries away the products of combustion. A liberal supply of available air is therefore very important. Yet in many cases the space allotted to the boiler room is inside, small and without adequate air supply for combustion. Boiler rooms should be of ample size and depth to accommodate the boilers without crowding, and should have an abundant supply of air for both combustion and ventilation. The space in front of the boilers should be ample for convenience and comfort. A cramped boiler room is not only unsightly, but it also adds to the difficulty of taking care of the plant efficiently. The attendant, when firing, has to stand about 4½ or 5 ft. from the front of the furnace and usually about 12 to 18 in. to the left of a straight line running through the centre of the furnace door. He should have ample room to swing his scoop from the coal pile into any part of the furnace.

Many a fireman is blamed for the poor economy shown by the plant he operates where the dissatisfaction should be charged at least partially to the plant designer. It is difficult to keep skillful firemen in a small, poorly-kept boiler room.

The size and type of boiler to be specified and the evaporation the boiler will give are problems in which the advice of the boiler maker may well be considered. The boiler maker is usually quite willing to co-operate if provided with such data as the total radiation in square feet and pounds of condensation, total condensation of the steam and return lines in equivalent square feet of radiation and pounds, the quality and size of fuel available, the size and height of chimney and the firing period to be allowed.

CHAPTER X

Selection of the Proper Type of Steam Heating System

THE heat requirements of the building having been determined, the next step is the selection of the proper type of steam heating system to fit the particular needs. It is essential that the system of supply and return piping shall be such that the circulation of steam will be positively and uniformly maintained and that the air and the products of condensation shall be disposed of continuously in order that the system shall be efficient as well as economical in operation.

Two broad types of two-pipe steam heating systems have proved so successful during the past 20 years that their use has become the modern standard practice.

Each type is flexible in its application and may be modified in detail to meet the variable conditions that arise.

These two types are the *Open Return-Line* or *Modulation System* and the *Vacuum System.*

In the *Open Return-Line* or *Modulation System* a pressure slightly above atmosphere is maintained in the supply piping and radiators, the products of condensation flowing by gravity to a point of disposal at which atmospheric or occasionally slightly lower pressure exists. Here the air is vented through suitable devices and the condensation is either returned to the boiler, if one is provided, or wasted to the sewer, if the source of supply is a so-called "street system."

In its simplest form a modulation system consists of a low-pressure steam boiler and its appurtenances, supply piping, radiating surfaces, a modulation or graduated control valve at the inlet of each radiator and a thermostatic return trap at the outlet, a system of return piping with a device at the end to automatically remove the air and return the water of condensation to the boiler. Under favorable conditions the boiler operates, after initial heating, for long periods under vapor or partial vacuum, but due to the flexibility of the system, higher pressures are permitted in severe weather, when maximum heating requirements exist. It is very important that the steam pressure shall be closely controlled by means of an extremely sensitive damper regulator which will maintain the pressure always within a few ounces of that for which the regulator is set, thus making it possible to operate the boiler at or near atmospheric pressure during mild weather. The damper regulator also serves to quickly check the fire whenever there is a tendency for the pressure to rise, due either to a sudden closing off of a considerable amount of the radiating surface or carelessness on the part of the attendant, after firing up the boiler.

For reasons of safety it is necessary that the device returning the condensation to the boiler shall function properly when the steam pressure

rises above the normal operating point and even when, for short period, it reaches the blowing-off pressure of the safety valve, which is ordinarily not over 10 lb. in an open return-line system.

In the *Vacuum System*, a pressure at or slightly above or below atmospheric is maintained in the supply piping and radiators, and air and the water resulting from condensation of steam are continuously removed by *mechanical* apparatus which maintains, in the return piping, a pressure less than atmospheric. The partial vacuum required to remove the air and condensation is produced and maintained by mechanical displacement of the vapors of condensation.

The two types of systems are similar, in that a positive circulation of steam is secured by the natural flow of the heating medium from a higher to a lower pressure. The distinguishing difference between the two types lies in the method of removing and disposing of the air and the products of condensation.

In either modulation or vacuum systems, modulation or graduated supply valves, when attached to the radiators permit control of the room temperature by simple hand operation, ensuring a distinct saving in fuel. The efficiency of either system is dependent to a large extent upon the ability of the return trap on the radiator to free it of all air and water of condensation without at the same time permitting the escape of any steam.

The open-return or modulation system finds its widest application in a building covering a moderate area, in which the steam requirements are for heating only and where the radiation can be placed high enough above the water line so that the condensation will flow by gravity to the boiler. The system is noiseless in operation, simple in design, requiring no power-driven apparatus and except for periodical firing of coal and removal of ashes, the attention required is negligible.

There are a number of modifications of the modulation system, depending upon varying conditions, and a system installed in a residence for instance, may be quite different from that in a hotel or school.

The special advantages of the vacuum system can be realized to the fullest extent in projects such as the following:

(a) A group of buildings scattered over a considerable area where savings in cost of installation can be effected by the use of smaller size supply and return piping.

(b) One or more buildings so located with respect to the boiler plant that lifts are necessary in the return piping.

(c) A plant utilizing the exhaust steam from the engines for heating purposes, wherein the elimination of the back pressure will save directly in fuel cost or permit the engine to do more work with the same expenditure of fuel.

The foregoing examples do not by any means cover the entire field for use, for the vacuum system can be used in numerous other types of buildings either as a regular vacuum system or in combination with the open return-line system. Indeed the adaptability of the two systems to widely different operating conditions makes possible the choice of one or the other for every type of building. In the following pages certain general rules

will be given which may influence the selection of a heating system for any particular case. Mention will also be made of modifications which may be desirable or necessary to suit individual conditions.

In determining which of these types to employ, experience is the best guide, as the building conditions present so many variable factors that it is impossible to cover the subject exhaustively within the space of this chapter.

When selecting a heating system, consideration should be given to the following points:

(a) Size and type of building.
(b) Use of building.
(c) Location of building and topography of site.
(d) Construction and architectural features of the building.
(e) Source of steam supply.
(f) Operation and attendance.

Size and Type of Building: The first point to consider is the size of the building and its type.

Residences: The prospective owner of a residence is particularly interested in the amount of attention necessary for operation and the economy of fuel. Whether he attends to the heating system himself or employs a caretaker, he desires a plant requiring minimum attendance.

The modulation system is the most suitable in every respect either for a 30-room house or for a small bungalow. Except for periodical feeding of coal and removal of ashes, the attention required by such a system is negligible. The ability to vary the boiler pressure through a range from the maximum permissible in very cold weather to a pressure at or slightly below atmosphere in mild weather, and to control the quantity of heat given off from each radiator by manipulating the graduated supply valve, result in a distinct economy. The heat emission and the coal consumption are regulated to correspond with the outside temperature and weather conditions.

Apartment Buildings: Apartment buildings are erected by the owner for the revenue which they will bring and a heating plant which can be operated with greatest fuel saving and the least janitor service is the best paying proposition. Unless the building spreads over too much ground or the overhead return piping cannot be properly graded without too much complication, the modulation system is particularly adaptable. The small amount of attention required by this system gives the janitor of the building more time for other duties. Control of the amount of steam admitted into each radiator gives the occupant of each room or apartment a convenient means of temperature regulation.

Store and Office Buildings: Where no mechanical system of heating and ventilation need be provided and where an open-return-line system can be applied, the same type of heating system can be used in the small store building as described for residences and apartments. This also applies with equal force to small and medium-sized buildings for offices and other commercial purposes.

Fig. 10-1. The entry of a modern apartment building showing heat outlets in the side walls

Public Buildings: In this classification may be included court houses, post offices, libraries, and schools of small type where the ventilating systems are of the indirect or direct-indirect gravity ventilation type. Such buildings have, as a rule, no other mechanical equipment besides the heating and ventilating plant. For these structures a modulation system with open-line return is recommended.

We have considered so far the type of building wherein the area is moderate, the steam requirement is for heating purposes only, the basement radiation is well above the water line of the boiler and the overhead return piping can be properly graded, as required in the open-line system. In such cases the simplest form of system can be installed, requiring a minimum amount of attention.

Frequently, however, conditions arise wherein the open-return piping cannot be run at a higher level than that of the water line of the boiler and discharge by gravity into the boiler, or where the radiation in the basement must be placed at or even below the boiler water line. The first mentioned situation occurs if the building covers considerable area or structural conditions cause the return piping to be kept well down from the ceiling. Where mechanical ventilation is installed having indirect radiation placed in the basement for warming the air, or where the character of the basement rooms is such that they will not be properly heated if the direct radiators are placed near the ceiling, it becomes necessary to locate them too low for successfully returning the water to the boilers by gravity. In such cases a vented receiver is installed and connected to either a motor-driven or steam-driven pump. The receiver contains a float at its water level, the rise and fall of which controls the operation of the electric motor or steam pump, and the water of condensation is automatically delivered to the boiler. The apparatus is placed at or below the floor on a suitable foundation and the water line of the heating system is thus governed by the level in the receiver, regardless of the water line of the boiler. Where no high-pressure steam is required for industrial or other purposes, the automatic return pump can be used in conjunction with low-pressure boilers, in which event the pump will have a motor drive.

Where high-pressure steam is required for various purposes, one or more high-pressure boilers are installed. Steam for heating is reduced to suitable pressure by means of a pressure-reducing valve and is circulated through the modulation heating system by gravity, returning to the vented receiver. In such cases, a steam-driven return pump is installed, taking steam at boiler pressure and discharging the exhaust into the heating system through a suitable oil separator. The condensation from the various pieces of apparatus utilizing high-pressure steam is also delivered to the vented receiver and thence returned to the boiler.

For buildings occupying considerable area and for groups of buildings to be heated from a common boiler plant, the vacuum system is to be preferred to the modulation system with vented receiver and return pump. Where lifts are necessary in the returns, the vacuum system is the best solution. In high buildings a vacuum system is usually selected, owing to the saving which can be effected by the reduced sizes of supply and return piping as well as radiator inlet valves and return traps. If high or medium-pressure steam is not required for any equipment or process, low-pressure boilers may be installed in connection with an electrically driven vacuum pump which will also return the condensation from the heating system to the boilers. From an operating standpoint a vacuum system with an electrically driven vacuum pump of the rotary type is quite as simple as the open-

return-line system or a modulation system, with a condensation pump.

Where the steam requirements for the building are such that high-pressure boilers are required, either motor-driven or steam-driven pumps may be used depending upon conditions which will be touched upon later.

Use of Building: The use of the building, or the portion of time during which the building is in use and must be heated, is an exceedingly important factor in the selection of a system.

Stores, office buildings, restaurants and the like are heated throughout during the daytime, while at night the requirement is only that of preventing the freezing of plumbing, water pipes, etc.

In school buildings the ventilating system is usually put into operation about 8 o'clock in the morning and shut down at 4 o'clock in the afternoon. Rural schools often do not have electricity available for driving the ventilating fans and in such cases a steam engine is installed for the purpose. The boilers are usually operated at about 30-lb. pressure, steam for heating purposes is reduced to 1-lb. pressure and the condensation is delivered to the boilers by an automatic return-pump and receiver. The exhaust steam from the engine and pump are utilized in the heating system after extraction of the oil by passing the steam through an oil separator.

Where electric current is available a combination modulation and vacuum system may be installed. In this case the boilers are operated at low pressure. While the mechanical ventilating system is in use, a motor-driven vacuum pump is employed to remove the air and the water of condensation from the heating system and discharge the water into the boilers. As soon as the ventilating system is shut down, the vacuum pump may be stopped and the direct heating system is then operated as an open-return line system, discharging the returns through a suitable trap, by gravity to the boiler. At night the heating plant requires almost no attention.

The heating of a theatre may be accomplished in very much the same manner. In this instance however, the ventilating system is in use in the afternoons and evenings, during which the plant is operated as a vacuum system. After the close of the night performance the change is made to a modulation system.

Heating systems in churches are usually operated intermittently. Where no mechanical ventilation is to be provided and where all radiation can be placed high enough above the water line for gravity return of the water of condensation to the boiler, a modulation system will give excellent results. It has the special advantage that by eliminating the use of wet returns, the danger of freezing of pipes, due to intermittent operation, can be avoided.

Heating systems in churches as a rule do not receive the best of attention and therefore the simpler the installation, the more satisfactory the service. The operation of the modulation system in draining the condensation back to the boiler entirely by gravity also avoids the slight noise that usually accompanies the action of mechanical devices, if the latter are employed for handling condensation. Where mechanical ventilation is installed in a church the combination of a modulation and vacuum system will be found to operate with the same reliability as described in connection with school buildings.

The use of the motor-driven vacuum pump will ensure a rapid as well as noiseless circulation of steam and quick removal of the air during the warming-up period. If it is not possible to eliminate the wet returns where the combination system is installed, care must be used to properly protect the pipes against freezing.

Hotels, hospitals, institutions, asylums and the like have a 24-hour period during the entire heating season. It is absolutely essential that the service shall be continuous. Not only must the system be economical and noiseless in its operation, but it must also be very flexible to meet the varying demands of outside temperatures and weather. The comfort of each individual must be considered, and as is well known, the preferences vary. In all the above cases there are demands for high and reduced-pressure steam for various purposes and it is therefore probable that high-pressure boilers will be installed.

With a single building covering a moderate area, an open-return-line system in conjunction with an automatic pump and return tank, together with modulation supply valves on the radiators, will meet all of the requirements outlined above. For a group of buildings or a single building covering considerable area, a vacuum system will be more flexible.

In addition to all of the benefits of the modulation system, the vacuum pump will circulate steam very rapidly through the system, which is an important factor when quick heating up becomes necessary. A further advantage is the saving effected through the ability, in mild weather, when the demand for steam is light, to distribute a small volume of steam throughout the entire system as needed. With the modulation supply valve on each radiator properly adjusted, or with the radiators controlled automatically by thermostats, rooms on the cold side of the building will receive the proper amount of heat and those on the warm side will not be overheated and all this is brought about with a relatively small amount of steam.

Location of Building and Topography of Site: Location of the building and the general topography of the site not only affect the type of heating system used but may also influence the kind of boiler selected.

For example, in rural districts electricity may not be available as a motive power or it may not be advisable on account of its unreliability. Where mechanical ventilation is required, as for example in a school or other similar building, a steam engine will be required for driving the fan. A type of boiler capable of generating steam at, say 25 to 30-lb. pressure must be selected. The steam for heating is reduced to 1-lb. pressure. A low-pressure steam pump will have to be installed to return the condensation to the boilers, operating in conjunction with a vented receiver and an automatic float control device. The receiver must be located a sufficient distance below the bottom of the indirect radiators so as to obtain the necessary fall required to secure a rapid flow of water by gravity. A modulation system of heating with open return line to receiver will give excellent results and if the direct radiators are provided with graduated supply valves, the quantity of heat given off by each may be easily controlled by hand.

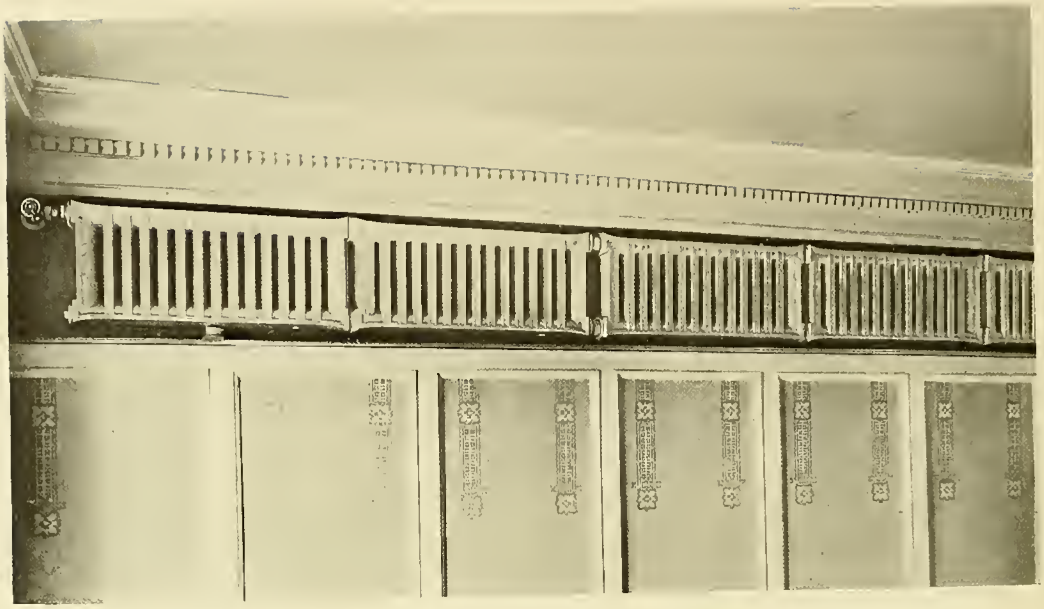


Fig. 10-2. Arrangement of cast-iron wall radiation in cove of ceiling in a grill room. This can also be employed in barber shops and other basement rooms where a modulation system is installed and it is necessary to keep the radiators well above the boiler water line. Operation of the steam inlet valves of such radiators can, if necessary, be facilitated by the use of extension stems or chain attachments

Fig. 10-3. Cast-iron wall radiation in garage. The radiation is placed at some distance from the floor level to avoid being damaged by cars and to prevent injury to tires from heat

The topography of the site may make the return of condensation difficult or impractical except by the use of vacuum or by direct pumping.

Where lifts are necessary in the return the vacuum system is the only solution.

A group of buildings spread out over considerable area, supplied with steam from a central boiler plant, may require the use of the vacuum system in order to balance the pressure differential between the supply and return at the several buildings, particularly if it is contemplated to add new buildings to the group at some future time. A further distinct advantage of a vacuum system under these conditions lies in the ability to use smaller pipe sizes for both supply and return lines, with a consequent reduction in cost of installation.

Construction and Architectural Features: The construction and architectural features of the building present a variety of problems.

It is frequently necessary to heat finished basement rooms by placing the radiators under the windows or at other points near the floor. Under these circumstances the condensation from the low radiators will not return to the boilers by gravity, as they are located at or below the water line and it becomes necessary to install a vented receiver and an automatic return pump or a vacuum pump. The former will operate in conjunction with a modulation system, the latter with a vacuum system.

Frequently the structural conditions of the building are such that the return piping has to be run near the ceiling of the basement or along the floor of the first story. The lifts resulting from this situation make the use of a vacuum system imperative. A very high building or one covering a very large area impose such conditions that the vacuum system is again the best solution of the problem. The greater pressure differential which results in this system, enables the use of smaller piping or with the same size piping reduces the back pressure which must be carried in the engines and pumps to secure complete circulation.

Reference has been made to the use of modulation supply valves to control the quantity of steam delivered to the individual radiators. In department stores, loft buildings, warehouses, factories, etc., where there are large open spaces to be heated, usually containing a number of radiators, the modulation supply valves may be omitted and a fair degree of temperature regulation may be obtained by completely shutting off one or more unit.

Sources of Steam Supply: There are three main sources of steam supply:

1. Live steam from high-pressure or low-pressure boilers.
2. Exhaust steam from engines, turbines or auxiliaries.
3. High-pressure or low-pressure steam from outside sources.

If steam is required for heating purposes only, the selection of the boiler will be a part of the heating problem, based upon the building requirements. Where no mechanical apparatus is necessary, low-pressure boilers will be the natural choice. If electric current is not available and the system requires fan engines, return pumps or vacuum pumps or other power-driven apparatus, a type of boiler should be selected which is capable of operating

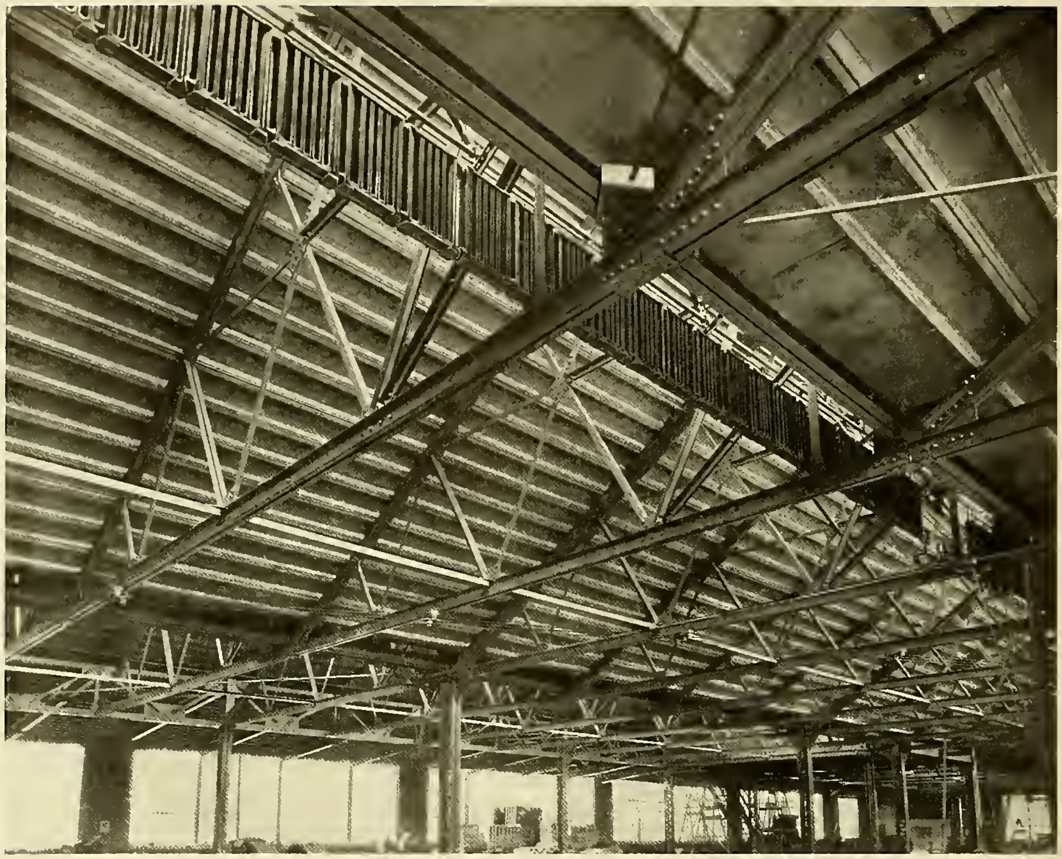


Fig. 10-4. Cast-iron wall radiation arranged under the saw tooth of a factory roof

Fig. 10-5. Arrangement of cast-iron wall radiation on side walls of a factory building

successfully with a working steam pressure of not less than 25 to 30 lb.

The conditions under which the heating boiler must operate are to a large extent the governing factor in its selection. The kind of fuel, the intensity of draft which can be obtained, the length of time between firing periods, the character of attention which the boiler will receive, the abuse which it will stand without injury, are all important factors which should receive due consideration.

The dimensions of the fire box, the proportion of direct and indirect heating surface in the boiler, the area and type of grate and the available draft are all influenced by the kind of fuel which is to be burned. This in turn depends upon the geographical location of the plant and the local fuel market. In many cities, boilers larger than a given size are required to comply with more or less drastic smoke prevention laws. It will thus be seen that the fuel plays a very important part in the choice of a boiler.

Sufficient attention is not always paid to selecting a boiler having flues and surfaces perfectly accessible for cleaning. It is also important in operating the boiler to see that there is a systematic removal of all soot and dirt at regular and frequent periods. In parts of the country where the water contains heavy scale-forming material, boilers having interior pockets should be avoided as scale will accumulate easily in these pockets and cannot be removed even by more frequent blowing down.

The shape of the boiler room may have some influence upon the type of boiler selected. A long, narrow room may lend itself better to the use of tubular or steel fire-box boilers, while a nearly square space may be best adapted to the cast-iron sectional pattern. Where tubular boilers are selected provision should be made for ample space to clean the tubes and to replace them, when renewals become necessary.

The question of future extensions should be considered when the problem is in the preliminary stage. Unless provisions are made then, the owner may find it very expensive to add to the boiler equipment at a later date.

In buildings requiring 24-hr. heat, such as hospitals and like institutions, reserve units should be installed to provide for possible breakdown.

In low-pressure installations, where a part or all of the water is returned by mechanical means, such as a motor-driven return pump, or a vacuum pump, it frequently happens that the water is delivered intermittently and in "slugs" instead of continuously; or it may fail completely on account of interruption of the current. The boiler must be of such a type or constructed of such material that it will not be injured by the sudden lowering of the water line, even to the dangerous point. In other words, the design or construction should be chosen which will permit the greatest withdrawal of water per inch drop in water level.

Priming of boilers arises from a number of causes, among which may be mentioned grease and dirt within the boiler, impurities in the water, lack of proper steam disengaging surface, insufficient steam space, and too high velocity of steam at the boiler outlets.

If a type of boiler likely to produce priming must be selected for physical reasons, the arrangement of the connecting piping must be such as to eliminate any possibility of trouble from this source.

The Architect must not lose sight of the fact that a boiler used solely for heating purposes lies idle, without a fire under it, for a period of from four to five months of each year, depending somewhat upon the latitude and length of the heating season. Recognition must be taken of this fact in selecting the kind of material which is best adapted to withstand the corrosive action which is likely to occur in a damp basement room. If the material is subject to rust and deterioration when lying idle, it should have additional thickness to offset this action.

It seems hardly necessary to call attention to the fact that boilers should conform to all requirements of local and State ordinances, and that compliance with the Boiler Code of the American Society of Mechanical Engineers will ensure first-class material and construction.

The designer of the plant for a residence is in most cases confronted with two conditions which decide the type of boiler which he shall use: *first,* smallness of the boiler room, and *second,* the low head room in the basement. Both suggest the use of the cast-iron type of boiler because of its compactness and low water line.

If there are other uses for steam, the type of boiler or the source of steam supply may be definitely fixed by other considerations than the requirements of the heating system.

Most large modern hotels in cities are provided with high-pressure boiler plants, either for generating their own electric power or, in case electric current is purchased, for operating the pumping equipment and furnishing steam for kitchen and laundry purposes. The vacuum system with steam-operated vacuum pumps is proper for heating such buildings.

Great progress has been made in recent years by the country towns in providing more convenient hotel accommodations for the traveling public. The owner of the small-town hotel, while not in a position to equip with all the refinements of a metropolitan hotel, is anxious to have his guests provided with comfortable and properly heated rooms and therefore wishes to install an efficient and economical plant. The modulation system either with gravity return or with vented receiver and return pump is particularly advantageous, giving all that can be asked in heating effect, and enabling the janitor or engineer, who is also, in many cases, the porter, bell boy and general utility man, to take care of his many other duties.

Y. M. C. A. buildings resemble the first mentioned type of hotels in many respects, as in addition to the recreational features, hotel accommodations are provided for the members. Restaurant and cafeteria service are maintained, as well as swimming pools, Turkish baths, etc., in connection with which there is a demand for high-pressure steam in addition to the low-pressure steam needed for heating. For this reason all the condensation cannot be returned directly to the boilers.

The heating system should be of a type which permits regulation of the supply of steam to the bedrooms, according to whether they are occupied or empty. The graduated control system of steam supply to the radiators by means of modulation supply valves is a logical system to adopt. A steam-operated pump and receiver takes care of the returns from all the steam-using equipment and also from the heating system.

The modern hospital has a considerable amount of steam-using equipment such as sterilizers, blanket warmers, steam cookers, dishwashing machines, laundry machinery, etc., requiring steam at pressures ranging from 30 to 90 lb. This makes a high-pressure boiler plant necessary. Many of the larger hospitals have their own electric power plants, and also use steam for operating refrigerating plants. In such cases the available exhaust steam should be utilized to the fullest extent and this is best accomplished by means of a vacuum system.

High-pressure boilers are usually installed in manufacturing plants where high-pressure steam is needed for process work and cheap electric power is not available. In such cases the necessary electrical machinery for generating current is installed and the exhaust steam is used for heating. As with hospitals and office buildings, the vacuum system ensures quick circulation of steam and removal of air and reduces the back pressure to a minimum. Where the plant extends over considerable area, the use of smaller size supply and return mains, and the ability to lift the condensation where changes of grade occur, become important factors.

In localities where street steam is available, with uninterrupted service guaranteed for the entire heating season, and where the rate does not exceed that at which steam can be generated in an individual plant, the installation of the modulation system with street steam supply provides very satisfactory heating for almost any type of building.

The reduced first cost of the heating plant, due to the omission of the boiler and its appurtenances, and the fact that such a plant requires practically no operating attention, make the arrangement very attractive from the owner's standpoint.

The service company supplying steam to the building usually extends the service pipe through the foundation wall and to this the heating contractor makes his connection. The water of condensation is discharged to the sewer through a meter in the return line, except where a flat rate per square foot of radiation is charged, in which case no meters are used.

This type of heating system can be installed in almost any type or size of building, except where too extensive area prevents satisfactory arrangement of the return line for gravity open-return circulation. In such cases, the motor-driven vacuum pump offers a simple means of insuring positive removal of the condensation and air.

Operation and Attention: The initial cost is frequently the deciding factor in the selection of a heating system, and it is not until the end of the first heating season, when the purchaser finds out the cost of fuel and caretaker's services, that the question of operation and attention receives the consideration to which it is entitled. In this chapter, however, it is not possible to more than touch briefly upon this important subject.

The most successful heating system is the one which will accomplish all of the results for which it is designed with the least amount of attention and the minimum expenditure for fuel. With the view of simplifying the system, the use of mechanical devices for handling the condensation should be limited to those cases where an open-line gravity return does not work

out satisfactorily. The conditions under which return pumps and vacuum pumps are necessary have been fully explained in previous paragraphs and it is not necessary to refer to the subject again.

We cannot emphasize too strongly the important part which the radiator return trap plays in the economy of operation. It should be of a type that will permit the rapid removal of all air and all condensation but at the same time prevent the escape of any steam. This point is explained very thoroughly in Chapter 14.

A system cannot be expected to give the best results unless all of the operating conditions are favorable. Three factors which have a great influence upon economical as well as successful operation are the location of the boiler room, its size and that of the chimney.

In planning the basement of any building the architect should pay particular attention to both the boiler room and the coal and ash storage spaces. It is needless to say that the coal room should be so placed that the labor of stowing away the fuel, and afterwards feeding it to the boiler, is reduced to a minimum, and that suitable means is provided for the economical handling of ashes.

Ample firing space must be provided in front of the boiler, ample room at the rear to give easy access to the return and blow-off piping and walking space at either side wide enough to enable the steamfitter to easily and quickly assemble the sections and later permit the application of the covering. If the boiler is the tubular type, there must be space for cleaning the tubes as well as for replacing them when repairs become necessary.

Limiting the depth of the boiler room is a false economy and will only result in partial if not almost complete failure of the heating system to give satisfaction. There must be sufficient grade so that the overhead return piping can be given ample pitch toward the boilers, thus ensuring quick return of the condensation by gravity, and so that the lowest point of the return for an open-return-line modulation system is at least 30 inches above the water line of the boiler.

Lack of head room reduces the pitch of the return piping to a minimum and narrows the selection of boilers to perhaps a single type, having a low water line but otherwise not at all adapted to the work which it must perform. It may also compel the construction of a pit, which is not always desirable, or require an electric return pump which may unnecessarily complicate a system that would otherwise be very simple.

Certain types of buildings require the simplest heating system possible. In residences the firing is infrequent and is done by the owner or a caretaker. The system must be rugged in design, with the least possible mechanical devices, but flexible enough to respond to varying changes of outside temperature and weather.

The janitors of school buildings have a multitude of duties to perform besides that of fireman. In the rural districts the school committees have limited appropriations for janitor service and apparatus has to be installed which is capable of giving satisfactory results with such unskilled attendance as is available.

As stated before, apartment houses are run on a business basis and the heating system must be economical of fuel and require little attention but must be flexible enough so that the occupants have a convenient and independent means of controlling the temperature of the various rooms. In the various types of buildings outlined above, the modulation system, with open return line to the boiler, will be found to meet the requirements of simplicity, flexibility and economy.

Passing to the combination of the open return line with either the automatic return pump and vented receiver or the vacuum pump, or the straight vacuum system, we find the same economy and flexibility with the addition of comparatively simple mechanical return apparatus.

Summing up the advantages of the modulation and vacuum system we find them to be as follows:

Modulation Systems:

1. Simple in design.
2. Efficient in fuel.
3. No expert attendance required.
4. Quick response to demands for changes in rate of heating.

Vacuum Systems: 1. Circulation of steam is quick, positive and uniform. All surfaces are heated in a relatively short space of time after steam is turned into the system.

2. Saving in operating cost is accomplished practically by eliminating back pressure upon steam engines. This either saves directly in fuel cost or permits the engines to do more work at same expenditure of fuel.

3. Saving is effected through the ability during mild weather, when demands for heating are slight, to distribute a relatively small volume of steam throughout the system as needed, with a pressure at or even slightly below the atmosphere. In this country, mild weather constitutes about 75 per cent of each heating season, moderately cold weather about 20 per cent and only 5 per cent can be classed as "severely" cold.

4. Saving of fuel results from utilizing the condensation and its contained heat as part of the boiler feed.

Certain advantages are common to both systems, as follows:

Modulation and Vacuum Systems: 1. Noiseless in operation. Water hammer is unheard of due to continuous relief of air and positive removal of condensation.

2. Radiators maintained at 100 per cent heating efficiency due to complete removal of air and water. Absence of air valves on radiators eliminates one of the most annoying features of many heating systems.

3. Independent temperature control of each room at the will of the occupant.

4. Efficient in fuel.

To the foregoing advantages should be added comfort and convenience. More and better work is obtained from occupants of properly heated buildings.

CHAPTER XI

Flow of Low-Pressure Steam Through Piping

FLOW OF STEAM THROUGH PIPES: Flow of steam through piping is caused by difference in pressure, which diminishes continually from the source to the outlet, due to frictional resistance, deflection, contraction and expansion. Likewise there is a continual drop in temperature due to the transmission of heat through the walls of the piping.

Steam at initial pressure and density, but without material velocity, as in a boiler, requires a certain pressure drop, to impart initial velocity in the main. This drop varies with the velocity required, density of steam and shape of the orifice at entrance of the main. The pressure drop or head required for a given velocity, as of initial density at a point about three diameters beyond the entrance of a steam main, with sharp entrance edge, has been found from tests of the weight of low-pressure steam passing through a cylindrical sharp-edged orifice of length equal to three diameters. The pressure difference or head (h_1) necessary to produce such velocity (v_1) is fully 1.7 times that found by the well known velocity formula, $v = \sqrt{2\ gh}$.

It seems reasonable to assume that a like pressure drop is necessary to impart initial velocity within the heating main from a boiler or steam drum, as contrasted with the exhaust of an engine, reducing valve, etc.

Table 11-1 gives 1.7 times the pressure drop or head (h_1) in pounds and ounces per square inch, based on the above assumption.

Table 11-1. Velocity of Steam in Feet per Minute Within Entrance of Main (as of Initial Density) Produced by Pressure Drop ($p_1 - p_2$) = h

From various absolute initial pressures in pounds per sq. inch = p_1

p_1-p_2 Ounces per sq. inch	p_1-p_2 Pounds per sq. inch	Velocity in feet per minute — Absolute initial pressure p_1: 15	16	17	18	19	20
	.01	2260	2203	2138	2086	2036	1980
¼	.0156	2830	2758	2665	2610	2544	2475
	.02	3200	3115	3020	2950	2880	2800
½	.0312	3995	3885	3770	3680	3595	3495
	.04	4530	4405	4270	4175	4070	3960
¾	.0468	4910	4775	4625	4520	4420	4280
	.05	5060	4930	4780	4660	4540	4420
1	.0625	5660	5520	5340	5220	5090	4950
	.07	6000	5840	5660	5520	5390	5240
1¼	.0781	6350	6180	5980	5840	5710	5540
	.08	6420	6240	6050	5910	5770	5610
	.09	6800	6615	6410	6260	6120	5940
1½	.0937	6940	6750	6540	6390	6250	6060
	.1	7170	6980	6760	6610	6460	6260
	.11	7520	7320	7090	6925	6770	6570
	.12	7860	7650	7420	7240	7060	6870
2	.125	8020	7810	7560	7390	7220	7010
	.13	8180	7960	7710	7520	7350	7150
	.14	8470	8250	7990	7800	7610	7410
	.15	8790	8560	8290	8090	7910	7680
2½	.1562	8970	8740	8460	8760	8080	7840

Friction in Run: Steam, having attained initial velocity at the entrance of the main by a pressure drop $(p_1 - p_2)$, will require a further drop $(p_2 - p_3)$ to overcome friction.

Various formulae have been published by which to determine the velocity or weight of steam of given quality which with a given pressure drop will flow in a given time through a given length of straight pipe of given uniform diameter.

Analysis of the principal formulae, after reduction to common terms, indicates a substantial agreement among the majority of these formulae n the following fundamentals:

that the velocity varies as the square root of the pressure drop.

that the velocity varies as $\frac{1}{\text{density}}$

that the pressure drop varies as $\frac{1}{\text{density}}$

that the pressure drop is proportional to length of run.

that the pressure drop varies as the square of the weight flowing.

The various fundamental equations for frictionless pipes may be reduced to the following form:

$$w = \sqrt{\frac{(p_2 - p_3)\frac{1}{s}d^5}{L}}$$

and the allowance made for friction by multiplying the radical by a constant or numerical value, dependent on the diameter, in the following form:

$$w = c\sqrt{\frac{(p_2 - p_3)\frac{1}{s}d^5}{xL \text{ or } (L + y)}}$$

in which

w = weight of steam flowing per minute.
c = a constant or numerical value
p_1 = absolute pressure of initial steam when quiescent.
p_2 = absolute pressure within entrance of main.
p_3 = absolute pressure near end of main.
d = diameter in inches.
L = length of run in feet.
x = a factor of L derived from some sub-formula.
y = a formula or sum to be added to L in the basic equation.
$\frac{1}{s}$ = mean weight of 1 cu. ft. of steam in pounds.

Regarding the value of c, the late Professor Kent made the following apt statement:

"The coefficient of friction according to different authorities varies according to laws about which they do not agree."

Investigation demonstrates that many of the laboratory experiments and tests of commercial pipe lines upon which the values of c, x and y have

been estimated were so made as to include the pressure drop necessary for initial velocity while in others this is not included. Other tests appear to have been made on but one or at best a very few different sizes of pipe and lengths of run.

Some authorities assume that the factor c (which includes all friction) is constant for all sizes of pipe irrespective of relation of perimeter to included area of cross section; in this respect differing materially from all the commonly accepted formulae for flow of water. These among themselves assign materially different constant values to c.

Other authorities assign values varying with diameter, thereby recognizing the proportionate relation of perimeter to cross-section and the influence of surface retardation on the flowing mass. The two principal investigators of the latter school do not differ materially in the values assigned to c although J. M. Spitzglass, in his analysis,* goes exhaustively into the frictional elements (skin friction due to rubbing of the fluid on the rough surface of pipe and internal friction due to relative motion of particles of fluid on each other) and deduces a formula which takes into consideration both the coefficient of friction and the relative capacity of pipes of various diameters together with experimental coefficients for the various fluids.

Gebhart in his analysis of this subject makes the following very practical statement:

"Notwithstanding the numerous investigations conducted on laboratory apparatus and on pipe lines under actual power plant conditions, there is no trustworthy rule for accurately determining the flow of steam in commercial piping."

Professor R. C. Carpenter in his investigations regarding flow of steam in pipes reaches the following conclusion:

"For practical conditions, it is rather better to have an allowance in pipes for an excess in friction than to have the reverse condition true."

From an extended experience in steam heating practice and installation, it seems a fair conclusion that in none of the published formulae is sufficient consideration given to the excess friction liable to be encountered due to reduction in area and frictional resistance due to the very usual neglect of the workmen to ream pipes true after cutting.

This excess friction is likely to increase as the proportion of perimeter to area increases and be a serious source of inaccuracy in the determination of flow in the smaller sizes of commercial piping, and pipes if inadequate, when once installed, will usually remain a source of trouble and discredit.

This has led to the use of a table in which the value of c in the formula $w = c\sqrt{\frac{(p_2 - p_3)\frac{1}{s}d^5}{L}}$ has been increased for the smaller sizes beyond that of any of the authorities above referred to.

These values of c and the flow table based thereon are offered as those

* *Flow of Fluids and Frictional Resistance in Pipes*, J. M. Spitzglass, *Armour Engineer*, March and May, 1917

found to be adequate in practice under any but the worst practical conditions to which it has been applied.

$$W = 60c\sqrt{\frac{(p_2 - p_3)\frac{1}{s}d^5}{L}} \qquad (\textit{Formula 11-1})$$

W = weight of steam in pounds per hour.

Diameter of pipe in inches	1″	$1\frac{1}{4}$″	$1\frac{1}{2}$″	2″	$2\frac{1}{2}$″	3″	$3\frac{1}{2}$″	4″	5″	6″	7″
Value of c	34	41	44	49	52	55.6	57	59	61	62.5	63.4

Diameter of pipe in inches	8″	9″	10″	12″	14″	16″	18″	20″
Value of c	64.2	64.8	65.4	66	65	65	65	65

This formula makes no allowance for drop due to initial velocity, condensation, or changes in direction or area of pipe.

Table 11-2, Pages 114-5, has been computed from Formula 11-1

For example, ascertain the pressure necessary to overcome friction in a 400-ft. run of 4-in. straight pipe when conveying 600 lb. of steam per hr. and p_2 is 16 lb. per sq. in. absolute.

$\left(\frac{600}{731}\right)^2 = (0.821)^2 = 0.674$ lb. pressure drop for 1000 ft. due to weight of steam other than tabulated; therefore, pressure drop for given length, or 400 feet is:

$$0.674\left(\frac{400}{1000}\right) = 0.270 \text{ lb. per sq. inch.}$$

Condensation Loss: Through the entire length of run, there is a further loss of pressure, due to radiation and condensation. This loss is least in well covered mains with still air, at high temperature. Condensation in long runs of small pipe frequently causes the greatest loss of weight and occasions large pressure drop.

Figure 11-1, Page 116, gives averages of condensation loss in bare and covered pipes for various differences between temperature of steam in pipe and air surrounding the pipe or its covering.

The following example is given to call attention to what is likely to happen if tabular steam values, for straight runs, be used to size mains supplying radiation through long runs of small pipe, even if the mains are well insulated. From Table 11-2 it will be seen that a $1\frac{1}{2}$-in. pipe with a friction loss of 1/10 pound per 100 ft. and an initial pressure of 16 lb. absolute will convey steam at an hourly rate of 55.1 lb. or 53250 B.t.u. per hour.

By inspection of Figure 11-1, we find that if the difference in temperature between steam in the pipe and air surrounding it is 150 deg. fahr. and the pipe has good insulation, there is transmitted through that covering about 25 B.t.u. per lin. ft. ($\frac{1}{2}$ sq. ft.), or 25000 B.t.u. per hour for 1000 ft. run. Therefore, about 60 per cent of the entering steam will be condensed.

Effect of Deflection, Contraction and Expansion: Mains are seldom straight cylindrical pipe from end to end. Normally there are elbows, valves, branch outlets, reductions in size, separators, expansion joints, etc.,

each adding to frictional resistance and causing pressure drop.

Although not technically accurate, it has been found convenient in estimating, to express these resistances in units of the additional length of run of straight pipe that would produce an equal effect. Table 11-3, which is believed to be conservative and likely to produce results well within the tolerance necessary in so complicated a subject, is figured upon this basis.

Fittings of different manufacturers vary in resistance in similar sizes and similar fittings vary in percentage of resistance. No very careful tests covering the entire range of flow of water, air and steam are available for data, but those that do exist have been studied in making up this table.

Pressure Drop: The necessity for pressure drop to create flow in heating systems is further explained in following pages. Modulation and vacuum systems differ in degree of this pressure drop rather than in principle.

Table 11-2. Weight of Steam Flowing Uniformly in One Hour Through Standard Straight Level Pipes 1000 ft. Long, with a Loss of 1 lb. per Sq. In., from Given Initial Pressure Within Inlet End

P_2 = absolute initial pressure within entrance of main. r = latent heat of steam at absolute initial pressure P_2. 1000 B.t.u. = thousands of B.t.u. contained in the entering steam. V = velocity of steam in feet per min. at initial density

Nominal size	Actual internal dia. in inches	Actual outside dia. in inches	Linear ft. per cu. ft. of internal volume	Actual inside area in sq. in.	Linear ft. per sq. ft. of external surface	Sq. ft. of ext. surface per linear ft.	Actual internal $(Dia.)^{5/2}$	Constant C in formula	P_2	15	16	17	18	19	20
									S	26.27	24.79	23.38	22.16	21.07	20.08
									$\frac{1}{S}$	.03806	.04042	.04277	.04512	.04746	.04980
									r	969.7	967.6	965.6	963.7	961.8	960
									Lb.	14.2	14.63	15.08	15.48	15.88	16.23
1″	1.049	1.315	167.5	.86	2.9	.315	1.13	34	1000 B.t.u.	13.7	14.15	14.5	14.9	15.27	15.6
									Vel. ft.-min.	1044	1001	983	955	934	908
									Lb.	33.9	34.92	35.92	36.90	37.86	38.80
1¼″	1.38	1,66	96.1	1.5	2.3	.434	2.235	41	1000 B.t.u.	32.8	33.75	34.7	35.5	36.4	37.3
									V	1428	1385	1344	1310	1276	1248
									Lb.	53.4	55.1	56.6	58.2	59.65	61.1
1½″	1.61	1.9	70.6	2.04	2.01	.497	3.28	44	1000 B.t.u.	51.7	53.25	54.6	56	57.3	58.7
									V	1650	1607	1554	1518	1478	1444
									Lb.	111.2	114.5	117.7	121	124.2	127
2″	2.067	2.375	42.9	3.36	1.61	.621	6.13	49	1000 B.t.u.	107.9	110.8	113.5	116.4	119.5	122
									V	2082	2022	1960	1906	1865	1820
									Lb.	181.1	189.2	195.2	201.5	205	211
2½″	2.469	2.875	30.15	4.78	1.33	.751	9.58	52	1000 B.t.u.	178	182.9	188.3	194	197	202.5
									V	2432	2353	2295	2240	2170	2130
									Lb.	339	349	359	369.5	378	387
3″	3.068	3.5	19.5	7.39	1.09	.991	16.47	55.6	1000 B.t.u.	328.5	337.5	347	356	363	372
									V	2890	2808	2725	2660	2595	2530
									Lb.	500	515	530	545	558	572
3½″	3.548	4	14.58	9.89	.955	1.046	23.7	57	1000 B.t.u.	485	498	512	524	537	549
									V	3190	3095	3010	2925	2860	2790
									Lb.	710	731	752	774	794	812
4″	4.026	4.5	11.3	12.73	.849	1.177	32.53	59	1000 B.t.u.	688	706	725	745	763	779
									V	3520	3415	3315	3235	3155	3075
									Lb.	1290	1328	1368	1405	1440	1475
5″	5.047	5.563	7.22	19.99	.686	1.457	57.17	61	1000 B.t.u.	1250	1284	1322	1350	1385	1420
									V	4070	3950	3835	3740	3640	3550
									Lb.	2092	2158	2218	2280	2340	2392
6″	6.065	6.625	4.99	28.89	.577	1.733	90.6	62.5	1000 B.t.u.	2025	2085	2140	2190	2250	2295
									V	4565	4440	4300	4195	4100	3980
									Lb.	3065	3155	3250	3340	3425	3506
7″	7.023	7.625	3.72	38.74	.501	2.	130.7	63.4	1000 B.t.u.	2970	3050	3140	3210	3290	3365
									V	5000	4845	4710	4580	4475	4360

Table 11-2—*Continued*

Nominal size	Actual internal Dia. in inches	Actual outside dia. in inches	Linear ft. per cu. ft. of internal volume	Actual inside area in sq. in.	Linear ft. per sq. ft. of external surface	Sq. ft. of ex. surface per linear ft.	Actual internal $(Dia.)^{\frac{5}{2}}$	Constant C in formula	P_2	15	16	17	18	19	20
									S	26.27	24.79	23 38	22.16	21 07	20 08
									$\frac{1}{S}$	03806	.04042	.04277	.04512	.04746	.04980
									r	969.7	967 6	965.6	963.7	961.8	960
8″	7.981	8.625	2.88	50.02	.443	2.257	180	64.2	Lb.	4275	4400	4530	4652	4775	4185
									1000 B.t.u.	4140	4260	4375	4480	4580	4690
									V	5380	5240	5080	4950	4830	4720
9″	8.941	9.625	2.29	62.72	.397	2.58	239	64.8	Lb.	5735	5900	6075	6240	6400	6560
									1000 B.t.u.	5560	5710	5860	6005	6150	6310
									V	5760	5600	5410	5290	5175	5050
10″	10.02	10.75	1.83	78.82	.355	2.82	317.7	65.4	Lb.	7680	7930	8145	8360	8600	8800
									1000 B.t.u.	7440	7660	7870	8010	8270	8450
									V	6150	5880	5790	5610	5530	5380
12″	12	12.75	1.27	113.1	.299	3.3	198.8	66	Lb.	12170	12530	12900	13270	13620	13940
									1000 B.t.u.	11750	12120	12460	12750	13100	13360
									V	6800	6600	6410	6230	6100	5940
14″	14.25	15	.904	159.5	.255	3.90	766.5	65	Lb.	18420	18970	19520	20080	20580	21100
									1000 B.t.u.	17850	18340	18820	19300	19780	20250
									V	7290	7060	6800	6700	6540	6400
16″	15.5	16	.765	188.3	.239	4 16	945.9	65	Lb.	22750	23110	24100	24800	25450	26100
									1000 B.t.u.	22050	22620	23250	23900	24450	25100
									V	7630	7410	7190	7020	6840	6660
18″	17.5	18	.601	240	.212	4.71	1281	65	Lb.	30850	31750	32700	33550	34450	35250
									1000 B.t.u.	29940	30700	31650	32300	33150	33850
									V	8100	7850	7650	7425	7270	7060
20″	19.5	20	.483	298	.191	5.23	1679	65	Lb.	40300	41600	42800	44000	45200	46250
									1000 B.t.u.	39200	40200	41300	42300	43400	44400
									V	8550	8310	8050	7850	7660	7475

The pressure drop for lengths other than 1000 ft. will be $\frac{L_1}{1000} \times$ the tabular pressure drop, where L_1 is the new length in feet, and the weight of steam discharged will be $\sqrt{\frac{1000}{L_1}} \times$ the discharge given above.

The pressure drop varies as $\frac{1}{\text{density}}$. The pressure drop varies as the $(\text{weight})^2$.

The weight of steam flowing varies as $\sqrt{\text{pressure drop}}$.

Table 11-3. Resistance of Fittings in Feet of Straight Pipe to be Added to Actual Length of Run

Size of pipe in inches	Gate valve	Long sweep ell run of standard tee	Medium sweep ell reduced run of tee	Standard ell much reduced tee	Angle valve	Short bend	Side outlet tee	Globe valve
	Length in feet to be added in run							
2	2	3	4	5	9	11	17	19
2½	3	4	5	7	12	15	21	26
3	3	5	6	10	16	19	27	33
3½	4	6	8	12	19	22	32	39
4	5	7	9	14	22	24	36	45
5	7	9	11	18	27	30	44	58
6	9	11	14	22	32	36	51	70
7	10	13	17	26	37	41	56	82
8	12	15	20	31	42	47	63	94
9	13	17	22	35	47	52	68	104
10	15	20	24	39	52	57	76	117
12	18	24	30	47	62	68	91	140
14	20	26	33	53	71	79	105	160

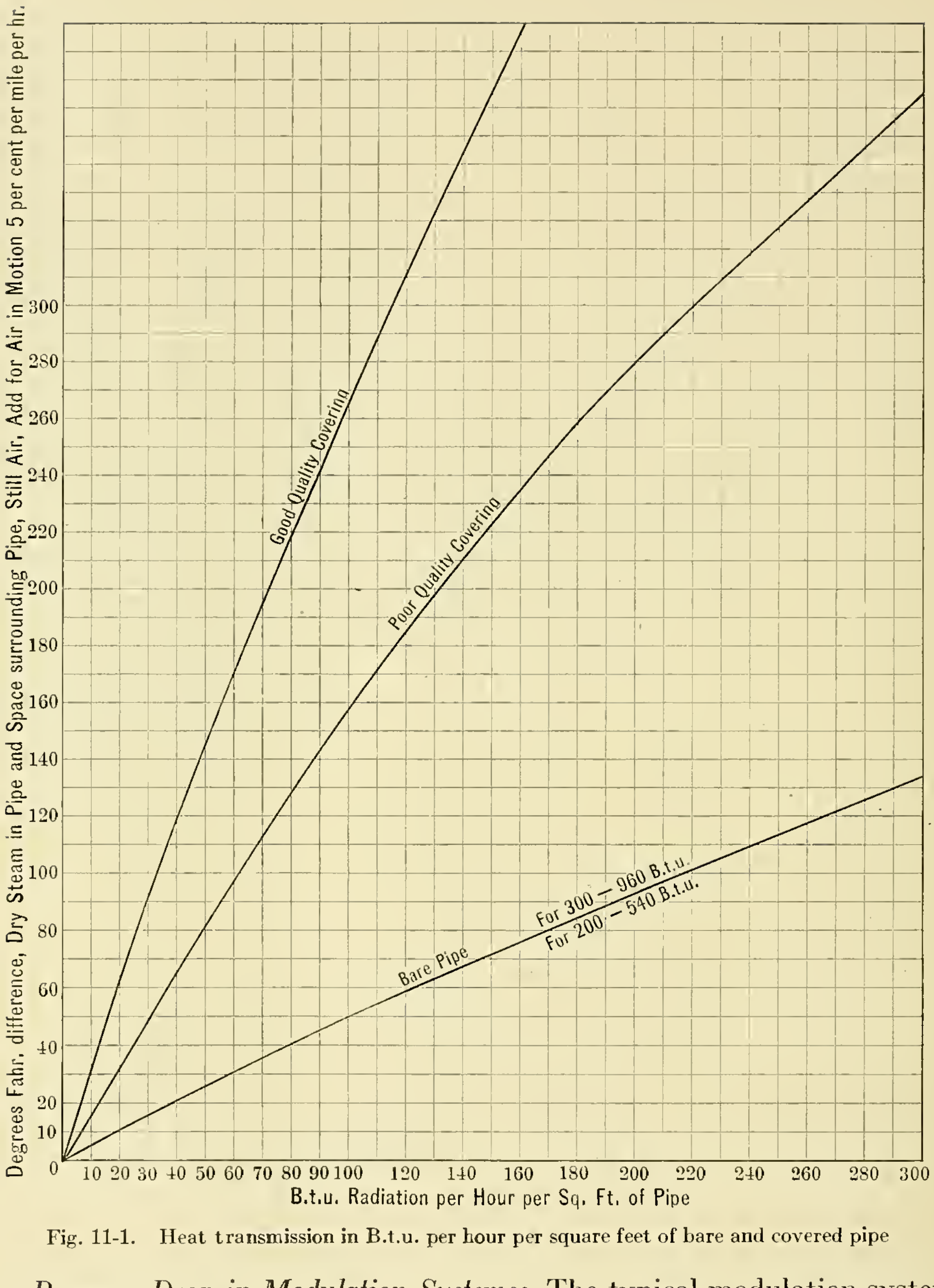


Fig. 11-1. Heat transmission in B.t.u. per hour per square feet of bare and covered pipe

Pressure Drop in Modulation Systems: The typical modulation system, as illustrated in Figure 11-2, when operating at normal rate, requires sufficient pressure against the valve piece of the vent valve p_1 to cause it to open against the atmospheric pressure. Representing atmospheric pressure as p and this excess pressure as p_1 the expressions $p + p_1$ = pressure at entrance of vent valve.

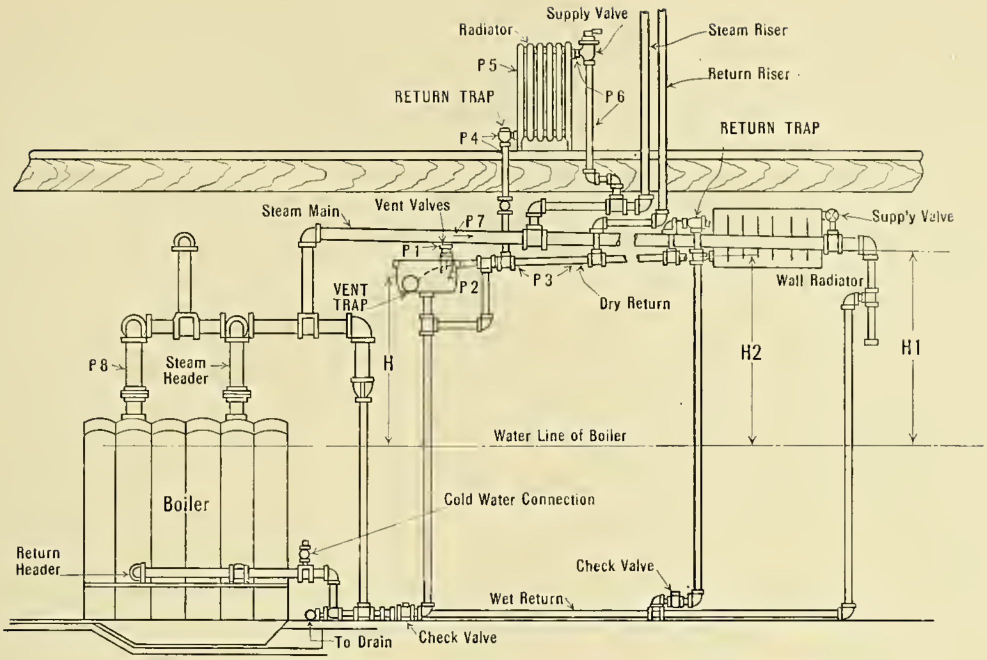


Fig. 11-2. Diagram of modulation system layout to illustrate pressure drop

To cause the air to flow from the vent trap through the vent valve orifice requires a pressure difference, which may be represented by p_2, varying with velocity of flow. Therefore, pressure in the vent trap becomes $= p + p_1 + p_2$. To cause the air to flow from outlet of the radiator trap through return main to the vent trap, there must be another pressure difference, represented by p_3, dependent on velocity of flow; also another pressure difference through orifice of radiator trap p_4. Therefore, pressure p_5 in the radiator at the time of air displacement by steam from the boiler must equal the sum of $p_4 + p_3 + p_2 + p_1 + p$. Of these last expressions p is relatively constant with gauge at zero lb. The flow through the vent valve p_1 is nearly constant, being mainly that pressure difference necessary to overcome the gravity of the valve piece, and adhesion of wet surfaces of the seat. The variable due to the volume of air passing is so slight, owing to low velocity, that it may usually be neglected.

The pressure p_1 of vent valve suitable for a modulation system is $\frac{1}{16}$ lb. per sq. in.

The pressure drop through vent valve orifice p_2 is a variable, greatest during initial heating-up period when a large volume of cool air is expelled from the heating system, and least during normal heating when velocity is that slight amount due to entrained air in condensation passing from the radiation. Air-vent traps are rated on basis of flow of initial air in 40 minutes in a system with $\frac{1}{16}$ lb. per sq. in. differential pressure through the vent-trap valve.

For less than rated capacity, either the time or pressure factor or both may be less; for instance, with p_2 constant, one-half the amount of radiation would require one-half the time period.

The pressure drop in the return main p_3 is also a variable, greatest during initial heating and dependent on length of run and maximum velocity. In a well-proportioned system, p_3 should never exceed 1/20 lb. per sq. in. differential between the farthest radiator trap and the vent trap, and during normal heating it is so slight as to be almost negligible.

The pressure drop through a radiator trap p_4 is also a variable, least and almost negligible during initial expulsion of air from radiation, at which time the trap-valve orifice is wide open. As the radiator warms up and condensation flows through the trap orifice with the last of the contained air, p_4 gradually becomes greater. It becomes maximum when condensation at or near steam temperature is flowing at the full rating of the return trap for a given p_4 of ⅛ lb. per sq. in., which pressure has been selected from tabular ratings of return traps (page 238). It is good practice not to have p_4 exceed ⅛ lb. per sq. in. where it is advisable to carry less than ½ lb. pressure on the boiler and ¼ lb. where a pressure of 1 or 2 lb. can be carried.

Representing the pressure difference necessary for flow, initially of air and subsequently of steam, from the radiator branch through the inlet or modulation valve to the radiator, requires another variable p_6, ⅛ lb. per sq. in. at full rating, least (in a properly designed modulation valve full open), during initial expulsion of air, and greatest when the valve is partly closed for modulation effect, at the selected rating of this valve, for a given pressure difference p_6.

p_7 is usually assumed for a system of mains, risers, branches and run-outs, designed from data on flow of steam in mains given in Table 11-8 to carry the maximum normal quantity of steam in a given time from the main heat pipe near the boiler to the inlet valve of farthest radiator, with this pressure drop p_7.

The quantity of steam, referred to in the preceding paragraph, flowing through the selected size main supply pipe will have velocity at the boiler which depends upon the pipe area and the volume of steam flowing in unit of time. To impart this velocity to the steam from a state of quiescence in the steam space of the boiler and to offset the resistance of the orifice requires another pressure drop p_8. Knowing the maximum normal quantity of steam and the size of the main, the pressure drop to give the resulting velocity can be obtained from Table 11-1.

It follows from consideration of the above that the pressure in the boiler P_b at time of maximum normal heating effect must be the sum of $p + p_1 + p_2$ etc., including p_8 as follows:

p, constant at atmospheric pressure.
p_1, at least intermittent at that time.
p_2, negligible at that time.
p_3, negligible at that time if return has proper grade.
p_4, tabular if full rated value in radiation is on farthest unit.
p_5, pressure drop in radiator, negligible at that time.
p_6, tabular if full rated value in radiation is on farthest unit.
p_7, from assumption in design from flow of steam in main. (Table 11-8).
p_8, that required for velocity head under above assumption. (Table 11-1).

The heating-up period will vary in accordance with initial pressure in the source of steam supply. Usually some time is required to raise steam to the normal pressure P_b. During that time air will be expelled and steam flow into the radiation at different rates due to the varying pressure caused by the increasing resistance of $p_1 + p_8$. If steam is constantly supplied during the heating-up period at pressure P_b, as when a central plant is the steam source, the condensation rate in the radiation due to absorption of heat by the metal will be as far in excess of normal as the sum of maximum $p_1 + p_2 + p_3 +$ an intermediate p_4, deducted from $P_b - p$, will produce a pressure difference (p_d) to cause initial velocity. It will flow through mains at a rate substantially in the same proportion as p_d is to p_7, provided initial velocity equal to p_8 has been previously imparted to the steam within the entrance of the main.

The intermediate p_4 referred to in the above paragraph is caused by the partial extension of the thermostatically moved valve piece in the return trap. This factor varies from full open and minimum resistance, when steam is first admitted and chilled condensation commences to pass, to nearly closed position and full resistance, when the radiator is completely filled up to the return trap with steam at a temperature corresponding with its pressure.

Modulation systems when operated at less than normal condensation may circulate continuously at pressure materially lower than the normal P_b, or may be intermittently operated at a pressure less than p, provided the air has first been expelled by a higher operating pressure. Under such conditions, however, the system will gradually become air-bound and cease to circulate.

In designing modulation systems, all gravity drip points should be provided with a hydraulic head (H_1) of at least 2½ feet for each pound per square inch of $p_7 + p_8 +$ frictional resistance in run of gravity drip and resistance of check valve between gravity drip and boiler, when the boiler is generating steam at its full capacity to supply cold radiation.

If the gravity drip be taken from radiation located below the dry return, with thermostatic air vent up to the dry return, then the resistance of any additional branch main, radiator, valve and check valve on gravity drip, must be added to $p_7 + p_6$, etc., given above, to determine whether H_2 is sufficient.

The hydraulic head in inches of water on the check valves will vary with the make, weight and angle of the clapper and the size of pipe tapping. This head is seldom less than 3 in. with the clapper at an angle of 10 deg. from vertical and may run up to 18 in. and higher with vertical-lift valve pieces.

In installing radiation with gravity drip for condensation as above, it is important that the branch connections and valve to such radiation have sufficient free area when in use, to cause little or no reduction in pressure in the radiator, from that in the main. A partially closed inlet valve might cause such a reduction in pressure, when added to the other resistances, that there would not remain sufficient total pressure in the radiator, when added to the available H_2, to overcome the pressure P_b plus the check valve resist-

ance in gravity drip. In consequence of this, condensation would build up in the column H_2, seal the radiator outlet and finally cause the radiator to become water-logged, possibly draining at a partial condensation rate, through the air vent into the dry return line.

The closing level of the air-vent trap should be located at such a height above the water line of the boiler that a hydraulic column is produced fully equal to the resistance of its check valve and drain pipe plus normal P_b.

This, however, is not as important as to have H_1 and H_2 ample. An air pressure will accumulate in this vent trap due to closing of the vent outlet, when column H is not sufficient to overcome resistance of drip line and the pressure P_b in the boiler. This air pressure will continue to build up with vent closed, until the built-up pressure with the assistance of column H overcomes the resistance of the boiler pressure. Then column H will fall, the air vent will open and allow escape of some air, thereby relieving part of pressure in the vent trap. Column H will again rise, closing the air vent, and this cycle will be repeated. When intermittent venting is repeated for a sufficient length of time under excess pressure without admitting raw feed water containing gases, all the air will be expelled from the radiation.

Such a system will continue indefinitely to circulate, due to a pressure difference which will be fully equal to that of its normal H; that is, the pressure in the vent trap will be less than the pressure in the boiler, by an amount equal to an hydraulic column of height H less the resistance of the check valve on the drip of this column.

In modulation systems designed for a stated pressure P_b and open vent at head H, the only difficulty occurs where a pressure exceeding P_b is built up rapidly before the initial air has been fully expelled. Under such conditions complete circulation will not be obtained as rapidly as if steam had been generated at a slower rate.

To overcome the difficulty in expelling air and returning condensation to the boilers, where excessive pressure is rapidly generated, as in the use of certain grades of bituminous coal, wood, etc., a special high-duty vent trap should be employed. In this trap, due to the rise in column H, the air vent is automatically closed and an equalizing pipe between the boiler and the vent trap is opened, the water under equalized pressure flowing by gravity to the boiler, after which the equalizing pipe is closed and the air vent again opened. The two operations taking place alternately, serve to vent the system completely of air and also return the condensation to the boiler, regardless of the boiler pressure.

As follows in the discussion of pressure drop in vacuum systems, the return mains should be proportioned relatively to the steam mains selected for equal duty. This principle applies also to modulation systems.

The basic proportional sizes of returns to supply mains recommended are given in Table 11-4.

Pressure Drop in Vacuum Systems: The reason for employing a vacuum system rather than a modulation system lies in the greater total drop obtainable from a given initial pressure P above, to terminal pressure p below atmospheric, thereby obtaining circulation through greater resistance due to long pipe runs and lack of grade for gravity flow of condensation.

Table 11-4. Relative Proportions of Steam Supply and Return Mains in Modulation Systems

Supply main	Dry return main	Return riser	Wet return
1	¾	¾	¾
1¼	1	¾	1
1½ and 2	1¼	1	1¼
2½	1½	1¼	1½
3 and 3½	1½	1½	1½
4	2	1½	1½
4½ and 5	2½	2	1½
6	3	2½	2
7 and 8	3 and 3½	3	2
9 and 10	4 and 4½	3½	2
12	5	4	2½

Lowering the terminal pressure p by mechanical exhaustion in return mains (the vacuum system) allows greater pressure drops through each of the series of resistance.

In good vacuum system practice, the total drop between source of supply through the inlet valve of the farthest radiator on the system should be that between available initial and atmospheric pressure, so that normally the pressure in the radiator will be at or very slightly below that of the atmosphere. The pressure drop p_4 of the return trap may usually be two to three times that permissible in a well designed modulation system. The drop p_3 in the vacuum return lines, if graded in direction of flow, may equal that in the supply mains of the system under consideration, and if it be necessary to elevate the condensation at one or more vertical lifts in order to obtain horizontal grade toward the vacuum pump, this (within limits of temperature of condensation) may be obtained by increasing the displacement of air and vapor by the pump. In systems where the high vacuum necessary to lift the condensation at one or more points, would occasion a needlessly high vacuum in that portion of return system which has a gravity flow, the degree of vacuum may be reduced by means of special vacuum controlling apparatus which provides for continuous discharge of condensation and also for a reduction of degree of vacuum between the inlet and outlet of the apparatus. (See Chapter 15, page 176, for description of such apparatus.)

In general, owing to greater pressure drop, a vacuum system will not require as large mains, branches to, and inlet valves of radiation as needed for a modulation system. Likewise, the radiator traps and return mains may be smaller for similar sized units of radiation provided radiator traps of high efficiency are properly installed to prevent leakage of steam to return lines.

Return traps on radiators should be proportioned for a pressure difference of between ½ and 1 lb. depending upon the condition of the particular problem.

Return mains should be proportioned relatively to the steam mains selected for equal duty by the table of comparative sizes (Table 11-5), allowing additional areas, however, where there is probability of high temperature in the outlet end of returns, due to steam leakage of return traps

or lack of vapor condensation occasioned by thoroughly insulated mains retaining the heat in the water passing through the radiator traps.

Where high vacuum for lifts increases the volume of vapors and gases to be removed, at least one size larger return mains should be used.

Such degree of partial vacuum should be carried by properly proportioned pump displacement as to cause a partial vacuum equal to the selected pressure difference (p_4) through the most remote return traps on the system. In proportioning pump displacement for vacuum systems, the most complex problem is that of proper allowance for the amount of vapor and air. Pressure below atmosphere in any part of the system is liable to induce invisible air leaks. For full efficiency of radiation, the temperature of condensation passing through return traps must be close to that due to the steam pressure in the radiator.

Part of the hot water, when flowing into lower pressure in the return line, flashes into vapor of high specific volume. The amount may be determined by inspection of the re-evaporation chart shown on page 157.

Some of this vapor will be condensed on the way to the vacuum pump, the volume depending upon whether or not the returns are insulated and also upon the amount of radiation, due to the length of the return pipe. It must be borne in mind that the vacuum or degree of partial pressure in the return line cannot exceed that corresponding to the temperature of the water of condensation.

Inleakage of air through even minute imperfections in piping causes an increase of volume to be handled proportionately as the absolute temperature of the air at inleak is to the absolute temperature in the return system, plus expansion from that volume at atmospheric pressure to that of vacuum pressure.

As explained in Chapter 13 on Vacuum Pumps, it is frequently possible to take advantage of some condensing medium such as cool air for ventilation, or water which must be warmed for cooking and washing, boiler feed, etc., and use this medium for cooling and condensing the air and vapor to decrease its volume on the way to the pump.

Table 11-5. Normal Relation of Return Mains and Risers to Supply Mains Caring for Equal Amounts of Steam in Vacuum Systems

Horizontal supply main	Horizontal return	Vertical return
1¼-in. and less	¾-in.	¾-in.
1½ and 2-in.	1	¾
2½-in.	1¼	1
3 and 3½-in.	1½	1¼
4, 4½ and 5-in.	2	1½
6 and 7-in.	2½	2
8 and 9-in.	3	2½
10-in.	3½	3
12-in.	4	3½
14 and 15-in.	4½	4
16 and 17-in.	5	4½
18 and 20-in.	6	5

Gravity drip vertical outlet at heel of risers 2½-in. and under, less than 12 stories high, ¾-in. Over 12 stories or over 2½-in. riser 1-in. vertical outlet increasing in horizontal run to 1¼-in.

Horizontal gravity drips graded at least ¼-in. in 10 feet are usually capable of caring for the number of ¾ or 1-in. outlets as follows:

Number of ¾ or 1-in. outlets which may be carried on one horizontal run when graded ¼-in. in 10 feet, provided radiation on steam riser does not drain as in one-pipe system

Size horizontal	No. of ¾-in. outlets	No. of 1-in. outlets
1¼-in.	12	6
1½	18	12
2	30	18
2½	60	36
3	100	50

Sizing of Piping: The use of the tables in sizing piping may best be explained by the following examples.

Vacuum System: Assume a central steam generating plant for a group of buildings, Figure 11-3.

In the problem here presented are a boiler house and three detached buildings A, B and C, connected by a system of well-covered mains in a tunnel. Through these mains it is desired to convey 6000 lb. of steam to building A, 5000 lb. to building B, and 3000 lb. to building C, per hour, with a pressure drop from 16-lb. absolute in the boilers to or near atmospheric pressure just beyond the main valve in each building.

Good covering, still air at about 70 deg. and proper drainage are assumed.

The total steam requirement per hour of buildings A, B and C is 14,000 lb.

The longest run of main is from the boiler house to building C and without allowance for fittings is 880 ft.

In estimating the sizes of pipes by the use of Table 11-2 it is necessary to first find the drop of pressure per 1000 ft. and then to find the corresponding quantity of steam flowing through the pipe for a drop in pressure of 1 lb. for this distance of 1000 ft.

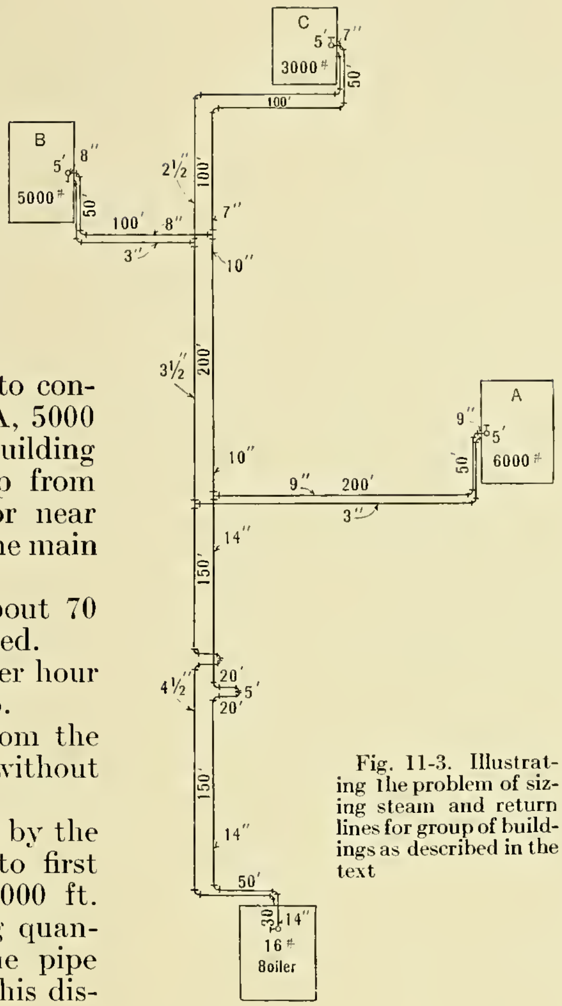


Fig. 11-3. Illustrating the problem of sizing steam and return lines for group of buildings as described in the text

The pressure drop varies directly as the length of a pipe, and the weight of steam discharged through a pipe varies directly as the square root of the pressure drop. We therefore multiply a given weight of steam by

$$\sqrt{\frac{\text{1 lb. x (the tabular pressure drop per 1000 ft.)}}{\text{The given pressure drop per 1000 ft.}}}$$

to find the equivalent weight of steam at 1-lb. drop per 1000 ft.

The assumed drop in pressure is 16 − 14.7 = 1.3 lb. per sq. in.

For a given total drop of 1.3 lb., the drop per 1000 ft. is

$$\frac{1.3}{880} \times 1000 = 1.48 \text{ lb.}$$

The first section of pipe to A conveys 14,000 lb. per hr. and the corresponding weight of steam at 1-lb. drop per 1000 ft. is

$$\sqrt{\frac{1}{1.48}} \times 14{,}000 = 11500 \text{ lb.}$$

Referring to Table 11-2 we find that to convey 11500 lb. per hr. at a pressure drop of 1 lb. per 1000 ft. requires a 12-in. pipe.

The second section of main from A to B conveys 8000 lb. per hr. at a pressure drop of 1.48 lb. per 1000 ft. Using the same reasoning we find that the corresponding weight of steam at 1-lb. drop per 1000 ft. is

$$\sqrt{\frac{1}{1.48}} \times 8000 = 6600 \text{ lb.}$$

From Table 11-2 the pipe size is found to be 10-in.

Similarly the branch from B to building C conveys 3000 lb. per hour at 1.48 lb. drop per 1000 ft.

The corresponding weight at 1 lb. drop per 1000 ft. is

$$\sqrt{\frac{1}{1.48}} \times 3000 = 2460 \text{ lb.}$$

And from Table 11-2 the pipe size is 7-in.

The total steam to be carried will, however, be in excess of 14000 lb. by the amount condensed in the mains.

The pressure drop for pipe friction will be less than 1.3 lb. by the amount necessary for initial velocity.

The length of run equivalent to the lineal run plus the added allowances for fittings as shown in Table 11-3 will be materially in excess of 880 ft.

It is therefore evident from an inspection of the plan that the above trial sizes may be too small and that it will be advisable to assume an increase of one size of pipe above those previously assumed, in all cases where there is a considerable number of fittings etc. This is true, in this problem, in the first section of the main.

The trial sizes will then be, 14-in. for the run to branch A; 10-in. to branch B and 7-in. to C.

Condensation Allowances: For 425 ft. of 14-in. main from the boiler to branch A.

From Table 11-2 we find the square feet of surface per lineal foot of pipe to be 3.9 sq. ft., this equals 1657.5 sq. ft. for the 425 ft. of 14-in. main. To this should be added 5 per cent for radiation from fittings making approximately 1740 sq. ft., radiating 50 B.t.u. per sq. ft. per hr., when the temperature drop is 216 deg. $-$ 70 deg. $=$ 146 deg. Multiplying 50 (B.t.u.) by 1740 (sq. ft.) and dividing by 968 gives the total condensation for the 14-in. main, which equals approximately 90 lb.

The 10-in. main condenses 2.82 (sq. ft. of surface per lineal ft.) $\times$ 200 (ft.) $\times$ 50 $\div$ 968 or 29.2 lb. $+$ 5 per cent for fittings $=$ 31 lb.

The 7-in. main condenses $2 \times 255 \times 50 \div 968$ or 26.4 lb. $+$ 5 per cent for fittings $=$ 28 lb.

It is evident that the condensation of the branches will be a small portion of the total quantity of steam carried by the main or branches. Estimating by comparison with branch to C, it is obvious that branches to A and B will condense hardly more than 40 lb. per hour each.

This gives us the total quantity of steam to be carried by the 14-in. main; $14000 + 90 + 31 + 28 + 40 + 40 = 14229$ lb. per hr.

The 10-in. main carries 8000 + 31 + 28 + 40 = 8099 lb. per hr.

The 7-in. branch to C carries 3000 + 28 = 3028 lb. per hr.

Pressure Drop for Initial Velocity: In a 14-in. main conveying 18970 lb. per hour the velocity is 7060 ft. per min., from Table 11-2. At 14229 lb. per hour the velocity will be $\frac{14229}{18970}$ x 7060 = 5300 ft. per min.

From Table 11-1 we find that 0.0625 lb. is required to accelerate the steam from rest in the boiler to a velocity of 5520 ft. per min. in the main. For 5300 ft. per min. the drop is therefore $\left(\frac{530}{552}\right)^2$ x 0.0625 lb. = 0.06 lb. drop velocity head. The residual pressure available for overcoming friction in the mains and branches is 1.3 − 0.06 = 1.24 lb. per sq. in.

Referring to Table 11-3 we find the equivalent resistance in feet of straight pipe to be added to the run for friction in fittings, etc. The various quantities are tabulated on page 126 and the summation of the quantities gives the equivalent length of pipe for each section and for the total. We find that the revised equivalent run is now 1559 ft. and with a given drop of 1.24 lb. in the total run, the drop per 1000 ft. is 0.796 lb. In the last column s found the revised actual pressure drop for each section.

The pressure drop through a pipe varies as the square of the weight flowing through it. If we know the weight of steam discharged through a pipe w th 1-lb. drop per 1000 ft. (as from Table 11-2) and wish to find the drop of some other weight (as the weights in column q on next page) we can obtain it by applying this law. The square of the quotient of the given weight divided by the tabular weight, times the tabular drop equals the drop for the given quantity (column s).

The total drop of 1.16 lb. shown in the table is as close to the desired drop as can be expected with commercial sizes of pipe. If the deviation had been greater, one or more of the trial sizes would have to be altered to bring the total drop nearer that desired. Inspection of column s will show in which portion or portions of the main the drop per 1000 ft. is fartherest from the average of 0.796 lb.

It is this section or sections that should be refigured.

The pressure available for friction in the branches is the difference between the total available drop of 1.24 lb. and the amount already utilized in the main up to the junction with the branch in question.

The procedure for determining branch sizes is exactly the same as for the mains; assuming one size larger than the calculated trial size, adding condensation and allowance for fittings and checking to see that the actual drop to the building is close to the permissible drop.

The drop in the branch to A is 1.24 lb. − 0.518 lb. = 0.722 lb. Dividing this by 255 ft. (actual length of run to A) × 1000 gives 2.83 lb. drop per 1000 ft. in this run.

The corresponding weight at 1-lb. drop per 1000 ft. is:

$\sqrt{\frac{1 \text{ (lb)}}{2.83 \text{(lb.)}}}$ x 6000 (lb.) = 3560 lb.; requiring (from Table 11-2) 8-in. pipe.

The drop in the branch to B is 1.24 lb. − 0.768 lb. = 0.472 lb. Dividing

Table 11-6

Section	Trial size pipe m	Actual length n	Equivalent length for fittings o	Total equivalent length (n+o) p	Weight of steam q	From Table 11-2 Weight passed by trial size at 1-lb. drop per 1000 ft. r	Actual pressure drop per 1000 ft. $\left(\frac{q}{r}\right)^2 \times 1$ (lb.) s	Actual drop in section of main $\frac{p}{1000} \times s$ t
Boiler house to branch A	14-in.	425	1 Gl. V. 160 ft. 6 ells 318 ft. 478 ft.	903 ft.	14229 lb. per hr.	18970 lb. per hr.	.564 lb.	.518 lb.
Branch A to branch B	10-in.	200	run of reducing tee 24 ft. Tot. 24 ft.	224 ft.	8099 lb. per hr.	7680 lb. per hr.	1.12 lb.	.25 lb
Branch B to bldg. C	7-in.	255	1 Gl. V. 82 ft. 3 ells 78 ft. run of reducing tee 17 ft. Tot. 177 ft.	432 ft.	3028 lb. per hr.	3155 lb. per hr.	.921 lb.	.398 lb.
		Total equiv. main 1559 ft.					Total drop	1.166 lb.

this by 155 times 1000 gives 3.04-lb. drop per 1000 ft. in this run. The corresponding weight is $\sqrt{\frac{1}{3.04}} \times 5000$ (lb.) = 2880 lb. requiring a 7-in. pipe.

Assume for the first trial, one size larger than figured above, to take care of the comparatively large number of fittings, etc. The branch to A will be 9-in. and to B, 8-in.

The estimated quantities of condensation are close enough for use in sizing these branches. The total quantity carried by branch to A is therefore 6000 + 40 = 6040 lb. and by branch to B, 3000 + 40 = 3040 lb.

Table 11-7

Section	Trial size pipe u	Fittings v	Total equivalent length w	Weight of steam x	Weight passed by trial size with 1-lb. drop per 1000 ft. y	Actual pressure drop per 1000 ft. $\left(\frac{x}{y}\right)^2 \times 1$ (lb.) z	Actual drop in branch w $\times$ z	Drop in main to branch	Total drop boiler house to bldgs.
Branch A	9-in.	br. tee 68 ft. 2 ells 70 ft. Gl. V. 104 ft. Total 242 ft.	497 ft.	6040 lb.	5900 lb.	1.04 lb.	.52 lb.	.518 lb.	1.038 lb.
Branch B	8-in.	br. tee 63 ft. 2 ells 62 ft. Gl. V. 94 ft. Total 219 ft.	374 ft.	5040 lb.	4400 lb.	1.31 lb.	.49 lb.	.768 lb.	1.258 lb.

Since the total drop from the boilerhouse to the building in each case is not far from 1.24 lb., or is at least as close as commercial sizes of pipe will allow, the trial sizes of 9-in. to A and 8-in. to B are correct.

The sizes of return mains should be based upon the sizes of the corresponding steam mains in the foregoing example.

By referring to Table 11-5 we find as follows: branch returns from buildings B and C are respectively 3-in. and 2½-in. to the junction, where they increase to 3½-in., continuing this size to the point where the 3-in. return from building A joins the above. Increase the return here to 4½-in. and continue this size to the vacuum pump in the boiler house.

Long computations such as the above are required only in connection with extensive distributing systems, where the cost of one size larger pipe becomes important.

For general use in sizing mains, branches and risers for both modulation and vacuum systems, Tables 11-8 A, B, C and D will be found sufficiently accurate if used with discretion. They are based upon 75 per cent of the values of Table 11-2 and will cover an ordinary amount of valves, fittings, etc., if globe valves are excluded.

In the use of Tables 11-8 A, B, C and D the permissible pressure drop between the inlet of the supply main and the farthest radiator determines the alphabetical sub-division of the table which is to be used. Table 23-7 in Chapter 23 gives a list of pressure differentials, which will be found reasonably accurate for various types of modulation and vacuum systems under ordinary conditions.

The following rules should be employed to determine which column of length of run should be used for horizontal or vertical pipes in the alphabetical sub-division selected.

1. For horizontal supply pipes, find the total run in feet along the pipe from the source to the farthest radiator and use the corresponding column in the table.

2. For sizing up-feed risers, add $\frac{8}{10}$ of the length of the vertical pipes to the total run found by Rule 1, and use corresponding column in table.

3. For sizing down-feed risers deduct $\frac{2}{10}$ of the length of the vertical pipes from the total run found by Rule 1, and use the corresponding column in the table.

4. The sizing of supply run-outs, especially those in which the condensation must flow by gravity in the opposite direction to the steam current, calls for special consideration and will be discussed in Chapter 12 on Critical Velocities in Radiator Run-outs.

5. The sizes of return mains and run-outs should be based on the sizes of supply mains, which will take care of a similar quantity, and are found by reference to Table 11-5. For convenience, the correct sizes of return mains and risers, for a given number of pounds of condensation, length of run and pressure differential, are also contained in Table 11-8 A, B, C and D.

Modulation System: In sizing piping for modulation systems, long computations such as described under vacuum systems are not necessary. The Tables 11-8 A to 8 D are sufficiently accurate for ordinary conditions.

Table 11-8. Ratings of Supply and Return Mains in Pounds of Steam per Hour, for Various Pressure Drops from Initial Pressure of 16 lb. Absolute, when in Horizontal Runs of from 300 to 1,000 ft.

These tables are found by taking 75 per cent of the values of straight pipe given in Table 11–2, to cover an ordinary number of valves and fittings, entrance velocity and other resistances to the flow of steam

A—1/8-lb. Drop in Pressure

Pipe sizes for modulation systems			Length of run in feet				
Return (from table 11-4)		Steam supply	300	400	500	750	1,000
Return riser	Dry return main		Rating in pounds of steam per hour				
3/4″	3/4″	1″	7.08	6.12	5.48	4.48	3.88
3/4″	1″	1¼″	16.9	14.6	13.1	10.7	9.25
1″	1¼″	1½″	26.6	23.	20.6	16.85	14.6
1″	1¼″	2″	55.4	47.8	42.9	35.	30.35
1¼″	1½″	2½″	91.5	79.	71.	57.8	50.2
1½″	1½″	3″	169.	146.	131.	107.	92.5
1½″	1½″	3½″	249.5	215.5	193.4	157.7	136.5
1½″	2″	4″	353.5	305.	274.	223.5	193.6
2″	2½″	5″	642.5	554.	498.	406.	352.
2½″	3″	6″	1043.	900.	808.	660.	572.
3″	3″	7″	1525.	1318.	1185.	965.	836.
3″	3½″	8″	2130.	1840.	1650.	1347.	1168.
3½″	4″	9″	2855.	2465.	2215.	1806.	1564.
3½″	4½″	10″	3835.	3315.	2975.	2425.	2110.
4″	5″	12″	6060.	5230.	4700.	3835.	3320.
5″	6″	14″	9175.	7920.	7120.	5800.	5030.
5″	6″	16″	11320.	9780.	8780.	7160.	6210.
9″	7″	18″	15350.	13280.	11900.	9720.	8410.
7″	8″	20″	20100.	17400.	15600.	12720.	11030.

B—1/4-lb. Drop in Pressure

Pipe sizes for modulation systems			Length of run in feet				
Return (from table 11-4)		Steam supply	300′	400′	500′	750′	1,000′
Return riser	Dry return riser		Rating in pounds of steam per hour				
3/4″	3/4″	1″	10.03	8.67	7.75	6.34	5.48
3/4″	1″	1¼″	23.9	20.7	18.5	15.1	13.1
1″	1¼″	1½″	37.7	32.6	29.2	23.8	20.6
1″	1¼″	2″	78.4	67.8	60.7	49.6	42.9
1¼″	1½″	2½″	129.5	112.	100.3	81.8	71.
1½″	1½″	3″	239.	207.	185.	151.2	131.
1½″	1½″	3½″	353.	305.5	273.	223.	193.4
1½″	2″	4″	510.	433.	387.	316.	274.
2″	2½″	5″	910.	786.	704.	574.	498.
2½″	3″	6″	1478.	1280.	1142.	933.	808.
3″	3″	7″	2160.	1870.	1670.	1365.	1185.
3″	3½″	8″	3015.	2610.	2335.	1905.	1650.
3½″	4″	9″	4010.	3495.	3125.	2550.	2215.
3½″	4½″	10″	5430.	4700.	4200.	3430.	2975.
4″	5″	12″	8580.	7420.	6640.	5430.	4700.
5″	6″	14″	13000.	11250.	10060.	8220.	7120.
5″	6″	16″	16050.	13890.	12400.	10130.	8780.
6″	7″	18″	21750.	18820.	16820.	13750.	11900.
7″	8″	20″	28500.	24650.	22050.	18000.	15600.

Table 11-8—*Continued*

C—½-lb. drop in pressure

Pipe sizes for modulation systems			Length of run in feet					Pipe sizes for vacuum systems		
Return (from table 11-4)		Steam supply	300′	400′	500′	750′	1000′	Steam supply	Return (from table 11-5)	
Return riser	Dry ret. main		Ratings in pounds of steam per hour						Horiz.	Vert.
¾″	¾″	1″	14.18	12.28	10.98	8.95	7.75	1″	¾″	¾″
¾″	1″	1¼″	33.8	29.25	26.2	21.35	18.5	1¼″	¾″	¾″
1″	1¼″	1½″	53.3	46.2	41.3	33.7	29.2	1½″	1″	¾″
1″	1¼″	2″	110.8	96.	85.8	70.3	60.7	2″	1″	¾″
1¼″	1½″	2½″	183.	158.6	142.	115.8	100.3	2½″	1¼″	1″
1½″	1½″	3″	338.	292.5	262.	213.5	185.	3″	1½″	1¼″
1½″	1½″	3½″	498.	432.	386.5	315.	273.	3½″	1½″	1¼″
1½″	2″	4″	707.	612.	548.	447.	387.	4″	2″	1½″
2″	2½″	5″	1285.	1113.	997.	813.	704.	5″	2″	1½″
2½″	3″	6″	2085.	1808.	1620.	1320.	1142.	6″	2½″	2″
3″	3″	7″	3050.	2645.	2365.	1930.	1670.	7″	2½″	2″
3″	3½″	8″	4260.	3690.	3300.	2695.	2335.	8″	3″	2½″
3½″	4″	9″	5720.	4945.	4430.	3610.	3125.	9″	3″	2½″
3½″	4½″	10″	7675.	6650.	5950.	4850.	4200.	10″	3½″	3″
4″	5″	12″	12130.	10500.	9400.	7660.	6640.	12″	4″	3½″
5″	6″	14″	18360.	15900.	14220.	11600.	10060.	14″	4½″	4″
5″	6″	16″	22620.	19610.	17560.	14310.	12400.	16″	5″	4½″
6″	7″	18″	30750.	26600.	23800.	19420.	16820.	18″	6″	5″
7″	8″	20″	40250.	34850.	31200.	25420.	22050.	20″	6″	5″

D—1-lb. drop in pressure

Pipe sizes for modulation systems			Length of run in feet					Pipe sizes for vacuum systems		
Return (from table 11-4)		Steam supply	300′	400′	500′	750′	1000′	Steam supply	Return (from table 11-5)	
Return riser	Dry ret. main		Ratings in pounds of steam per hour						Horiz.	Vert.
¾″	¾″	1″	20.2	17.4	15.5	12.65	10.98	1″	¾″	¾″
¾″	1″	1¼″	47.8	41.4	37.	30.2	26.2	1¼″	¾″	¾″
1″	1¼″	1½″	75.4	65.4	58.3	47.6	41.3	1½″	1″	¾″
1″	1¼″	2″	157.	136.	121.5	99.	85.8	2″	1″	¾″
1¼″	1½″	2½″	259.	225.	202.	163.6	142.	2½″	1¼″	1″
1½″	1½″	3″	478.	414.	370	302.	262.	3″	1½″	1¼″
1½″	1½″	3½″	706.	612.	546.	446.	386.5	3½″	1½″	1¼″
1½″	2″	4″	1000.	867.	774.	632.	548.	4″	2″	1½″
2″	2½″	5″	1820.	1575.	1410.	1150.	997.	5″	2″	1½″
2½″	3″	6″	2955.	2565.	2282.	1867.	1620.	6″	2½″	2″
3″	3″	7″	4320.	3750.	3340.	2730.	2365.	7″	2½″	2″
3″	3½″	8″	6030.	5230.	4660.	3810.	3300.	8″	3″	2½″
3½″	4″	9″	8080.	7020.	6250.	5110.	4430.	9″	3″	2½″
3½″	4½″	10″	10870.	9425.	8400.	6860.	5950.	10″	3½″	3″
4″	5″	12″	17160.	14900.	13300.	10850.	9400.	12″	4″	3½″
5″	6″	14″	26000.	22550.	20130.	16400.	14220.	14″	4½″	4″
5″	6″	16″	32050.	27810.	24800.	20250.	17560.	16″	5″	4½″
6″	7″	18″	43500.	37720.	33620.	27450.	23800.	18″	6″	5″
7″	8″	20″	57000.	49100.	44100.	36000.	31200.	20″	6″	5″

The total quantity of steam to be supplied per hour at the time of maximum normal heating effect being a known factor and the total maximum pressure drop in the heating system being determined for this period, the pressure drop in the supply main must be so chosen that the pressure to be carried on the boiler will exceed by a safe margin the sum total of resistances between the boiler and the outlet of the vent valve.

For an illustration, assume a typical modulation system which requires 500 lb. of steam per hour for maximum normal heating effect. The length of run is assumed to be 300 ft. and the boiler pressure is not to exceed ½-lb. gauge.

To find the proper size of supply main to meet these conditions, the pressure drops from p to p_6 as described in the discussion of pressure drop in modulation systems, Page 116, must be determined, before the permissible pressure drop p_7 in the supply main can be ascertained.

During maximum normal heating effect we find the pressure drop from p to p_6 to be as follows:

p =	constant at atmospheric pressure =	0.000-lb.	gauge
p_1 =	pressure drop through vent check valve (intermittent at that period) = 1/20 lb. =	0.050 "	"
p_2 =	pressure drop through vent valve orifice (negligible at that time) =	0.000 "	"
p_3 =	pressure drop in return main. Negligible if return has proper grade =	0.000 "	"
p_4 =	pressure drop through orifice of radiator trap, which for the given condition will be the maximum tabular value of ⅛ lb. =	0.125 "	"
p_5 =	pressure drop through radiator. Negligible at that time =	0.000 "	"
p_6 =	pressure drop through radiator valve will be the maximum tabular value for the given period, ⅛ lb. =	0.125 "	"
	Total drop p to p_6 =	0.300 "	"
	The pressure to be carried on the boiler = ½ lb.	0.500 "	"
	Pressure drop p to p_6 =	0.300 "	"
	Difference of pressure available	0.200 "	"

Bearing in mind that in addition to the pressure drop p_7 in the supply main, we must consider also the pressure drop p_8 to impart initial velocity, we readily see that a pressure drop of ¼ lb. in the supply main would be unsafe and we, therefore, select the ⅛-lb. drop in the supply main p_7 as the basis for determining the size of pipe required.

We find by referring to Table 11-8 A that a 5-in. main is necessary to supply 500 lb. of steam with ⅛-lb. drop in pressure in a run of 300 ft.

We now have to determine the head or pressure drop p_8 necessary to impart initial velocity to the steam.

From Table 11-2, we find S, the cubic ft. per pound of steam at 15.3 lb. absolute (assumed boiler pressure) is very nearly 26.27.

Converting the total steam required in pounds per hour into cubic feet per minute

$$\frac{500 \times 26.27}{60} = \frac{13135}{60} = 218.9, \text{ or, say, } 219 \text{ cu. ft.}$$

By referring to Table 11-2, column 3, we find the linear feet per cubic foot volume, which for a 5-in. pipe is 7.22.

Multiplying 219 by 7.22 we obtain the velocity in feet per minute of the steam to be . . . . 1582 ft.

We now determine the pressure drop p_8 necessary to impart initial velocity and by referring to Table 11-1 we find for a 2500-ft. velocity, a pressure drop of 0.01 lb., which for a 1582-ft. velocity would be approximately 0.009 lb. per sq. inch.

The total pressure drop between the boiler and the outlet of the vent valve then becomes:

Pressure drop $p - p_6$ as stated before = ...	0.300-lb. gauge
Pressure drop p_7 in main ⅛ lb. = ...	0.125 " "
Pressure drop p_8 to impart initial velocity = ...	0.009 " "
Total pressure drop = ...	0.434-lb. gauge

We find an effective differential in pressure between the boiler pressure and the pressure losses in the sytem of $0.500 - 0.434 = 0.066$ lb. gauge, for maintaining circulation in the system during the period of maximum heating effect.

This proves that for the above condition, the ⅛-lb. drop in pressure in p_7 is the proper basis for selecting the table to be used, and this being determined, the intermediate sizes of the main and branches are taken from same.

The sizing of run-outs requires special consideration as described in detail in Chapter 12, Critical Velocities in Radiator Run-outs.

The sizing of returns involves the same procedure with modulation systems as outlined before in the discussion of sizing of piping for vacuum systems. The size of the return depends on the size of supply for an equal duty. By referring to Table 11-4, we find that the size of return corresponding to a 5-in. supply main is 2½ in., which is the size we select.

Taking care of the condensation in the steam main at the far point is often found necessary in modulation systems in which case the pipe sizes must be increased toward the end of the run, beyond the tabular values, to take care of the reduction in effective area of the pipe due to the condensation being carried along with the steam.

A further reason for increasing the sizes of the pipes toward the end of the run is to compensate for the air carried along with the steam in the pipes, which, if not properly relieved, will retard the circulation of steam to a great extent.

Air relief connections must be provided at the ends of the runs, through thermostatically actuated return traps into the nearest dry return, in all cases where gravity drips are made into a wet drip line.

CHAPTER XII

Critical Velocities in Radiator Run-outs

THE velocity in a nearly horizontal pipe in which the condensation is to be drained by gravity in the opposite direction to the flow of steam above it, becomes *critical*, when it reaches such rate that any velocity increase will cause the condensation to be swept upgrade against gravity. The need has been apparent to heating engineers of definite information regarding this critical velocity of steam in branch run-outs to radiation in which condensation must be drained in a direction opposite to steam flow.

Individual opinion based on experience regarding velocity permissible at given slope without danger of noise due to surging, varies fully 300 per cent.

Many modern buildings have very limited space in which to run pipes between the finished floor and the main beams and fireproof construction. There are many valid objections to exposing the run-outs above the finished floors, and the question frequently arises as to the proper size and grade for such pipes in the available space beneath the finished floor.

Fundamentally the size of pipe for a given radiator run-out is dependent on the maximum number of heat units to be conveyed in a given time. The latent heat content per cubic foot of steam at the range of pressures usual in modern "low-pressure" heating is least at the lower pressures. Denser steam at higher pressures undoubtedly sets up greater wave-forming friction of steam over surface of the condensation and will sweep the water up the slope at a slightly lower steam velocity than that at which the condensation will flow against the current of less dense steam. These facts in a measure offset each other and the small error in the final result will have less effect on the problem than the inaccuracies of grade liable to exist despite any reasonable care in erection.

In an endeavor to fix the critical velocity, a carefully conducted series of tests has been made. The first of this series was with glass tubes, to determine visually just what took place when steam at various velocities passed over its condensation in pipes graded against the steam flow. The result of this series was very instructive in determining the effect of velocity and what to look out for in subsequent tests. The second tests were with commercial pipe of 1 in., 1¼ in., 1½ in. and 2 in. sizes, each 18 ft. long; each pipe being tested at uniform grades of ¼ in., ½ in., 1 in. and 1½ in. in 10 ft.

It was found that the difference in critical velocity in the various sizes of pipe under test differed less at the same slope than the errors incidental to careful observation. In consequence of the fact that the size of pipe had no direct relation to critical velocity, only one size was tested at a grade of 3 in. in 10 ft. to complete the curve of velocity at slope.

The result of these tests upset some preconceived theories and established some facts that appear to be fundamental. These established facts are:

1. That the size of the pipe has no visible relation to the critical velocity, which was practically the same in all sizes tested.

2. That the normal volume of condensation in a covered pipe as compared with an uncovered pipe, had no effect on the critical velocity. In fact, increase in condensation up to the point where the volume of water limited the free area for steam and made a material difference in velocity, the condensation continued to flow as with normal condensation.

3. That greater or less length of run if at uniform slope makes no material difference. The controlling velocity is that in the first foot or two of pipe, and if the velocity existing there is above critical, it will sweep the condensation to the high end. In fact, increase in condensation up to the point where the volume of water limited the free area for steam and made a material difference in velocity, caused no change in flow of condensation.

4. That the direction of flow in the vertical supply riser to which the run-out is connected, will have a slight effect on the critical velocity in the run-out. The critical velocity is lower in a down-feed than in an up-feed riser. This is due to the change in direction of the highest velocity steam striking the run-out on the lower side and acting on the condensation which is endeavoring to flow in the opposite direction.

The most surprising fact demonstrated during these tests was the rapidly diminishing effect of a slope greater than 1 in 120 on critical velocity, and the indication from the curves plotted for the entire series, that the critical velocity was little, if any, greater at slopes of more than 1 in 40 than at that slope. It follows from the above that a velocity of steam which will sweep up the condensation in a pipe having a grade of 1 in, say, 33 will sweep the condensation upward in a pipe having more grade.

The practical application of this series of tests must take local conditions into consideration.

The thermal capacity of the mass of iron in a cold radiator, will call for a large volume of steam during the heating-up period, and at the same time the difference in pressure at the two ends of the run-out will be greatest. Consequently the velocity of steam through the run-out will be far greater

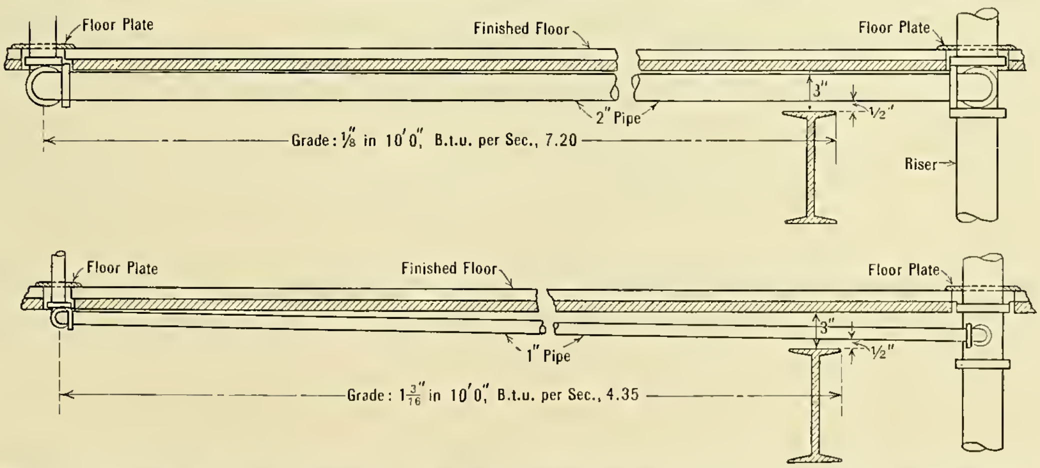


Fig. 12-1. Illustrating greater capacity of largest possible run-out pipe at a minimum grade compared with that of smaller pipe at much greater grade. The capacity of 2-in. pipe at grade of 1/8 in. in 10 ft. is greater than that of a 1-in. pipe at 1 3/8-in. grade in 10 ft. in the ratio of 7.20 to 4.35. Note application in limited space where run-out must cross structural frame beam

Fig. 12-2. Critical velocities in feet per minute, of low-pressure steam in radiator run-outs at various grades, where condensation flows down-grade against steam. Specific volume of steam, about 26.5 cu. ft. per lb.

during initial heating-up than during normal maintenance.

It is during the initial heating-up that the gurgling and hammering of condensation in run-outs causes most complaint. It is then that the flow of steam is most liable to exceed the critical velocity and sweep the condensation up into the vertical riser pipe to the inlet valve.

It would be possible to use a run-out of half the area of cross-section if the radiator is to be constantly hot during the heating season as compared with area of run-out at same grade for a radiator in which there are frequent alternations of heating and cooling. Again, there are many installations in which a little noise during the heating-up period would not be considered objectionable, while in others the same amount and kind of noise would condemn the entire heating system. No fixed rule based on square feet of radiation may therefore be made for sizing run-outs in which the condensation is normally drained against the flow of steam.

A few things are evident from these tests and a number must be left to the good judgment of the designer of the system under consideration.

Among the evident things are:

1. That a uniform grade approximating 1 in. in 10 ft. is about the maximum useful limit. That a pipe if uniformly graded when cold is liable to

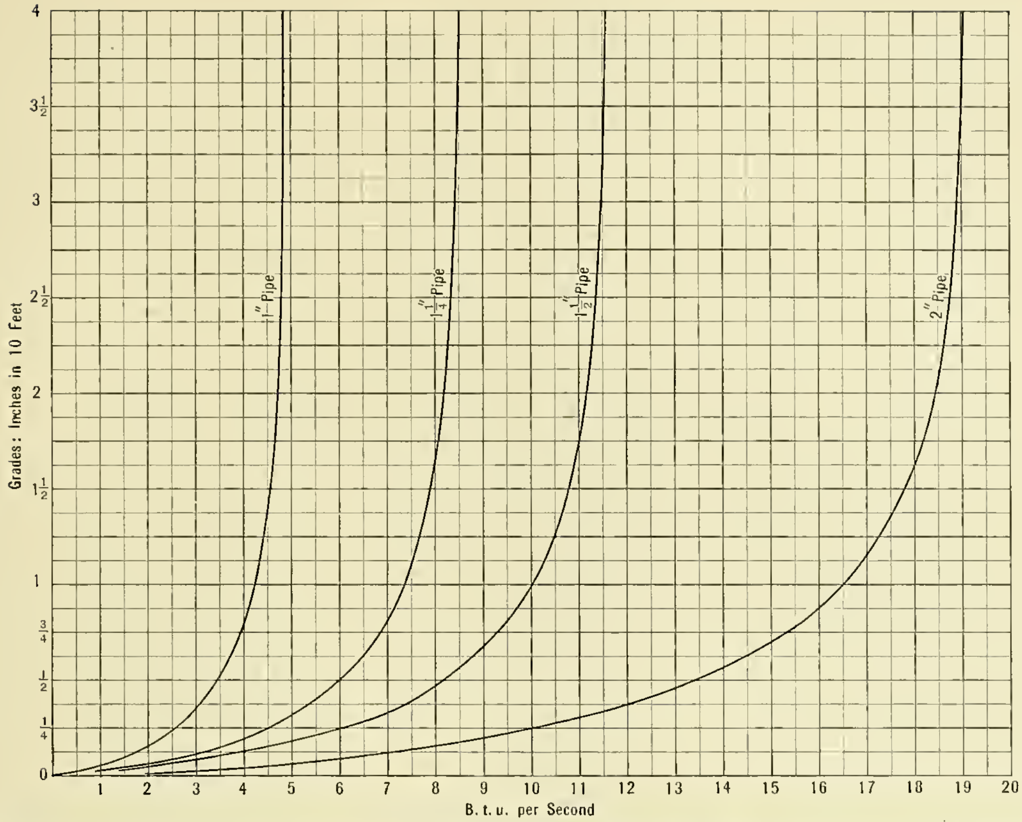


Fig. 12-3. B.t.u. per second conveyed in low-pressure steam through radiator run-outs at grades which are critical where condensation flows against the current of steam. Critical velocities established by test and as shown in Figure 12-2.

buckle upward in the middle when hot and destroy the uniformity of grade.

2. That the most constant annoyance will occur when the flow of steam, at normal maintenance rate exceeds the critical velocity for the grade at which the run-out is laid.

3. That where noise is permissible during the heating-up period, the run-out should be sized and graded so as not to exceed the critical velocity during any normal heat maintenance. If so sized there will be little if any noise during the initial period when condensation is being swept on into the radiator by a velocity materially in excess of about 1350 ft. per min. There will, however, be a considerable noise as the heat capacity of the metal in the radiator becomes satisfied and this will continue during the time the steam flow is at a velocity of about 1350 ft. per min. until the steam flow falls below the critical velocity at the grade of the run-out.

From the above tests certain practical conclusions may be inferred.

The practice in sizing run-outs has been based on some relation to pressure drop or the friction of the steam in the pipe. This more properly applies to mains and risers.

The pressure drop due to friction in any normal run-out, when velocity is low enough to permit the current of condensation to flow against the steam, is less than .001 lb. per ft., therefore so slight that it is negligible.

It would be much more consistent to size run-outs on basis of critical flow rather than on pressure drop.

Tables 1 and 2, based on the following assumptions, may prove of interest:

1. That a slight noise due to condensation flowing into the radiator with the steam during the heating-up period will not be objectionable.

2. That at maintained rate, the condensation in the vertical rise pipe must also flow back against the steam. This is not necessary where bottom of the inlet to the radiator is at a higher level than that of the outlet.

3. That the radiation during maintenance does not condense at a rate in excess of 250 B.t.u. per sq. ft. per hour.

4. That there will be a uniform grade of not less than 3/8 in. in 10 ft. in two-pipe connection and 1 in. in 10 ft. in one-pipe connection.

Table 12-1. Run-outs for Two-pipe Work Having Grade of Not Less Than 3/8 in. in 10 ft. Radiator Transmits Not More Than 250 B.t.u. per Sq. Ft. per Hour at Maintained Rate.

Size of pipe	1″	1¼″	1½″	2″
	Maximum radiation on pipe in sq. ft.			
Horizontal run-out grade 3/8 in. in 10 ft	43	72	101	173
Vertical branch and valve	58	108	114	260

Fig. 12-2. Run-outs for One-pipe Work Having Grade of Not Less Than 1 in. in 10 ft. Radiator Transmits Not More Than 250 B.t.u. per Sq. Ft. per Hour at Maintained Rate.

Size of pipe	1″	1¼″	1½″	2″
	Maximum radiation on pipe in sq. ft.			
Horizontal run-out grade 1 in. in 10 ft	25	50	68	115
Vertical branch and valve	35	75	100	170

CHAPTER XIII

Vacuum Pumps and Auxiliary Equipment

VACUUM PUMPS are used:

1. To remove air and other products of condensation from the return main where these products cannot be expelled to atmosphere by gravity or internal steam pressure alone.

2. To induce circulation by reducing the pressure in the return main, thereby increasing the pressure differential.

3. To assist in the complete disposal of the products of condensation.

Experience indicates two successful types of pump for this service, namely, reciprocating steam-driven, and rotating electric-driven. The steam-driven pump has efficiency and economy in its favor where steam at 30-lb. or greater, absolute pressure, is continuously available and the pump exhaust and its contained heat may be fully utilized in the system. The electric-driven pump is generally most efficient where exhaust steam from the engines and other sources is continuously available in greater quantity than is necessary to supply the heating system; in other words, where the exhaust from the vacuum pump to waste would be a loss. The electric-driven pump is also preferable where the available live steam supply has a pressure too low to operate a steam-driven pump.

Many rotating pumps in which both air and water were handled in one chamber have deteriorated very rapidly in service, largely because of the grit always present in the condensation. Rotating pumps with one pump chamber handling air and vapor and another containing a centrifugal impeller for handling the water have proved practical.

Many variables enter the problem of ascertaining the proper size of pump for a given heating system. In the final analysis, good judgment based on wide experience in applying a table of probable pump displacement is of far greater value than any theoretical formula.

Even for a close approximation, it is necessary to know enough about the heating plan in addition to "the square feet of equivalent radiation" to be able to estimate the probable maximum volumes in unit of time of both water and elastic fluids of condensation, necessary degree of vacuum at the pump and discharge head against which condensation must be delivered.

The volume of water-condensation varies in different installations fully 40 per cent per square foot of equivalent direct radiation. The volume of elastic fluids—air, water, vapor, steam and gases from impurities—also varies with the initial and terminal pressures, with the efficiency of the radiator traps, with the degree of prevention of inward leakage of air, with the probable cooling effect in the return, and with the character of the impurities in the boiler-feed.

Lifts (see Figure 13-1) in the return call for greater terminal vacuum with consequent greater expansion in volume of the elastic fluids, thus calling for greater pump displacement. They should, therefore, be avoided if possible.

Discharge head on reciprocating pumps handling water and air has the

Table 13-1. Data on Vacuum Pumps and Their Auxiliary Equipment

Diameter in inches Water cylinder	Condensation Pounds per hour	Condensation Thousands of B.t.u. per hour	Size of steam supply and vacuum governor for 75 to 100-lb. steam pressure	Minimum size of return main and of suction strainer	Minimum size of discharge to open tank	Sq. ft. water area required in tank for air separation	Size of plain or hydro-pneumatic tank: Diameter	Size of plain or hydro-pneumatic tank: Length	Size of tanks having water or steam control: Diameter	Size of tanks having water or steam control: Length
No. 1	No. 2		No. 3	No. 4	No. 5	No. 6	No. 7	No. 8	No. 9	No. 10
3"	510	494.7	½"	1¼"	¾"	0.33	4"	12"	18"	24"
4"	1047	1015.6	½"	1½"	1"	0.50	6"	12"	18"	36"
5"	1830	1775	½"	2"	1¼"	0.86	8"	18"	18"	36"
6"	2890	2803	½"	2½"	1½"	1.37	8"	24"	18"	48"
7"	4250	4123	¾"	2½" 3½"	1½" 2"	2.0	12"	24"	24" 24"	36" 48"
8"	5920	5742	¾"	3" 3½"	2"	3.0	12"	36"	24"	48"
9"	7980	7739	¾"	3½" 4"	2"	4.0	18"	30"	24" 24"	60" 72"
10"	10350	10010	1"	4" 4½"	2½"	5.0	18"	48"	30"	48"
12"	16300	15811	1"	4½" 5"	2½"	8.0	24"	48"	30"	60"
14"	24000	23280	1¼"	5" 6"	3"	11.4	30" 24"	48" 72"	36" 36"	60" 72"
16"	33500	32495	1¼"	6" 7"	3½"	16.0	36" 36"	60" 72"	36" 42"	96" 72"
18"	45000	43650	1½"	7"	4"	21.4	42" 36"	60" 96"	42"	96"
20"	58500	56745	1½"	7" 8"	4½"	28.0	42" 42"	72" 96"	48" 48"	72" 96"
22"	74300	72071	2"	8"	4½"	35.0	48"	96"	48"	96"
24"	92300	89531	2"	8" 10"	5"	44.0	48"	120"		
26"	112800	109416	2½"	10"	6"	56.0	60"	120"		
28"	135800	131726	2½"	12"	6"	65.0	72"	120"		
30"	161300	156461	2½"	12"	6"	77.0	96"	120"		
32"	189600	183912	3"	14"	7"	90.0	96"	120"		
34"	221000	214370	3"	14"	7"	105.0	96"	144"		
36"	254000	246380	3"	14"	8"	121.0	96"	144"		

Pumps having unequal stroke and bore

Stroke / Bore	Capacity factor
No. 11	No. 12
2.50	1.58
2.25	1.48
2.00	1.38
1.90	1.34
1.80	1.31
1.75	1.29
1.70	1.27
1.67	1.25
1.60	1.23
1.50	1.19
1.40	1.15
1.33	1.13
1.30	1.12
1.25	1.10
1.20	1.08
1.10	1.01
1.00	1.00
0.90	0.96
0.80	0.91
0.75	0.89
0.70	0.87
0.67	0.85
0.60	0.82
0.50	0.78

Standard conditions

(A) Steam above atmosphere at farthest radiator
(B) Units of 20 to 25 sq. ft.
(C) Standard screw-down radiator inlet valves
(D) No discharge near vacuum pump of large volume of condensation at steam temperature
(E) No lifting of returns
(F) Returns insulated
(G) Run of mains less than 500 ft.
(H) All drip points bave Webster Sylphon or No. 7 Traps

Factors for conditions other than above but Webster Traps used	Per cent. standard.	Equiv't to standard
	No. 13	No. 14
Long runs of uncovered returns	50	2.00
Blower stacks and water heaters in large units	66	1.50
120-sq. ft. units, returns covered 90 " " " " exposed	70	1.43
90 " " " " covered 70 " " " " exposed	75	1.33
70 " " " " covered 55 " " " " exposed	80	1.25
55 " " " " covered 40 " " " " exposed	84	1.20
40 " " " " covered 20 or 25-sq. ft. " " exposed	89	1.12
20 or 25-sq. ft. " " covered	100	1.00
10-sq. ft. " " exposed	111	0.90
Water seal traps, large units	125	0.80
" " " 80-sq. ft. units	143	0.70
" " " 40 " " "	166	0.60
" " " 15 " " "	200	0.50

Factors for approximate reduction of square feet of various types of radiation to pounds of condensation per hour

Factory pipe coils	0.33
Factory wall radiation	0.30
Low cast-iron radiators, 1 and 2 column	0.27
Medium height cast-iron radiators, 1 to 3-column	0.25
High cast-iron radiators, 2 to 4 column	0.23

effect of increasing clearance and slip and thereby decreasing the effective displacement.

A discharge head of more than one added atmosphere on reciprocating pumps is best handled by separating the water and gases and removing them independently through two separate pumps.

For slip in reciprocating wet-vacuum pumps it is seldom safe to allow less than ⅙ of the displacement, although a newly packed pump may show much less.

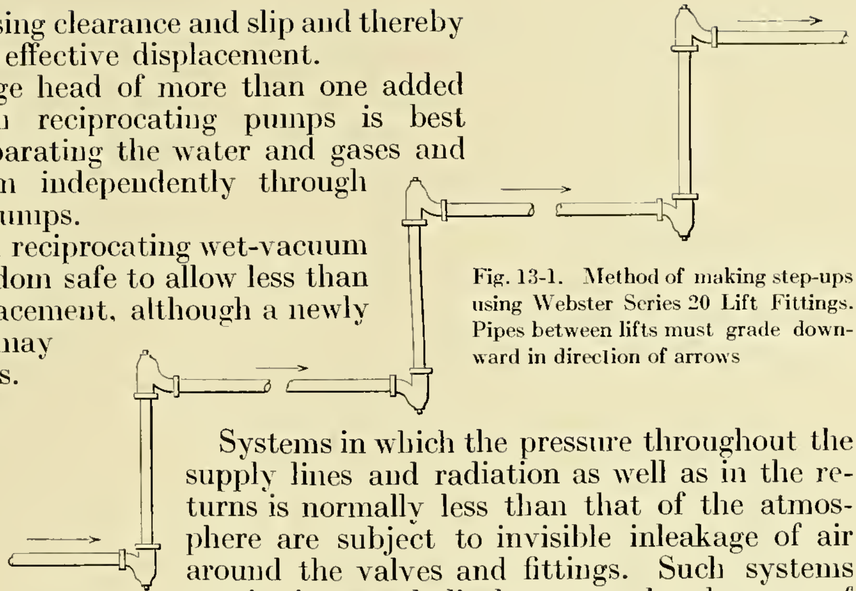


Fig. 13-1. Method of making step-ups using Webster Series 20 Lift Fittings. Pipes between lifts must grade downward in direction of arrows

Systems in which the pressure throughout the supply lines and radiation as well as in the returns is normally less than that of the atmosphere are subject to invisible inleakage of air around the valves and fittings. Such systems require increased displacement also, because of the greater volume of elastic fluids due to low terminal pressure necessary for circulation.

Cooling and consequent reduction in volume of the elastic fluids in the return present an element of considerable magnitude and uncertainty.

Well-insulated return pipes, also large volumes of condensation entering the main return close to the vacuum pump, require greater displacement than would the same radiation with returns in which a considerable portion of the vapors could condense between the radiation and the pump.

Clearance reduces effective displacement in all pumps. The clearance for a given cylinder diameter in reciprocating pumps of some makes is approximately the same in short-stroke as in long-stroke pumps. Commercial sizes of reciprocating vacuum pumps vary in ratio of bore to stroke between 1 to ¾ and 1 to 2; it follows that a pump of the latter proportion has greater efficiency per displacement than the short-stroke pump because of smaller percentage of clearance.

Experience with reciprocating steam-driven vacuum pumps indicates that for most favorable conditions the use of water cylinders of less displacement than eight times the normal volume of water of condensation is seldom safe. With radiation divided into small units, a ratio of at least 10 to 1 will be required.

Ratings for the rotating combination units should be based substantially on a 10 to 1 ratio of the combined displacement of water and air cylinders, the ratio of these cylinders to each other being about 2 of water displacement to 8 of air. In these pumps the displacement of water must be high on account of the constant speed, while a lower proportion of air displacement may be taken because of the high efficiency of the air chamber as compared with reciprocating pump cylinders which have greater clearance.

The speed and displacement in rotating pumps are normally constant, unless expensive variable speed motors are used, whereas in reciprocating

steam-driven pumps piston speed may be varied through wide range. The temptation to gain displacement by excessive piston speed without regard to the consequent racking of the pump because of too frequent starting and stopping of pistons and valves, should be avoided by adhering to a definite relation between piston speed and length of stroke. This relation as used for the calculation of basic ratings expressed in column 2, Table 13–1 is that the permissible piston speed in feet per minute equals 20 times the square root of the stroke in inches. These ratings are calculated for pumps having equal stroke and bore but they may be assigned to other pumps as will be explained later. The relation between the volume of the cylinder and that of the water discharged per stroke is figured as ten to one. Slip is assumed to be one-sixth of the total stroke.

The following example will fully explain the method of calculations for column 2.

Selecting from column 1, a pump having 4-in. stroke and 4-in. bore, the area of its cylinder is 4 x 4 x 0.7854 = 12.568 sq. in., or 0.0873 sq. ft.

The piston speed is 20 x $\sqrt{4}$ = 40 ft. per min., or 2400 ft. per hr.

The gross displacement is therefore 0.0873 x 2400 = 209.52 cu. ft. per hr., of which the gross water displacement is one-tenth, or 20.95 cu. ft. per hr.

Since the condensation will weigh 60 lb. per cu. ft. at about 200 deg. fahr., the gross water displacement may be expressed as 20.95 x 60 = 1257 lb. per hr. This must be reduced one-sixth because of slip, or to 1047 lb. of condensation per hr. as a basic rating for this pump.

Taking the average B.t.u. per pound of condensation as 970, the basic rating for the same pump may also be expressed as 1,015,600 B.t.u. per hr.

Ratings for vacuum pumps are properly expressed only in terms of pounds of water condensed by the heating system in a given period of time, or the equivalent latent heat in B.t.u. given up by the steam while condensing. Ratings in terms of "square feet of direct radiation" are not strictly correct and may be misleading since there is not recognition of steam pressures, temperature difference, and other factors entering the problem. However, for convenient use, factors are shown at the lower left of Table 13–1 for reduction of square feet of various types of radiation to pounds of condensation per hour which will give approximate results.

Since many vacuum pumps may have unequal stroke and bore, the capacity factors in column 12 are provided to show the relative effectiveness of such pumps as compared with "square" pumps having same bore and equal stroke. Column 11 shows relative proportions of "unequal" pumps in terms of stroke divided by bore. The corresponding factor for a pump of any selected relation of stroke to bore is found directly across in column 12.

These factors provide means for selection of stock size pumps where the rate of condensation to be handled is intermediate between basic rates for "square" pumps stated in column 2.

For instance, assume a condensation rate of 15,000 lb. per hr. To find the proper size of pump, select the diameter of bore in column 1 corresponding to the basic rate in column 2 nearest equal to the required rate. This

basic rate is 16,300 lb. per hr. and the bore is 12-in. Then find the factor in column 12 equal to the quotient of required rate divided by the basic rate for the 12-in. pump. This quotient is 0.92 and the nearest equivalent factor in column 12 is 0.91. The corresponding figure in column 11 is 0.80 which is the decimal relation of stroke divided by bore.

Multiply the bore (12-in.) by the factor (0.80) and it is found that the stroke should be 9.6-in. The nearest equivalent stock size of pump has 10-in. stroke and therefore a 12-in. x 10-in. pump is selected.

Where the result of such a calculation does not fit obtainable stock sizes, select a stock pump of some other diameter and stroke which, when factored by use of column 12, will give a rating at least equal to that required.

Another problem is that of finding the basic rating for any given pump of unequal stroke and bore; for instance, one having 4-in. bore and 6-in. stroke.

The relation of stroke to bore is 6 divided by 4 or 1.5. Finding the number 1.5 in column 11 it is noted that the corresponding factor in column 12 is 1.19. Multiplying 1.19 by 1047, which is the basic rating for a 4-in. x 4-in. pump from column 2, the product gives 1246 lb. of condensation per hr. as the basic rating for this 4-in. x 6-in. pump.

It is to be specially noted that the basic ratings shown in column 2 are calculated and shown for the standard conditions of operation stated in the upper right of the table. Other actual or expected conditions of operation can be transformed to terms of standard. Where the B.t.u. to be emitted are individually calculated for each group of like class and size of radiation, these quantities may be multiplied by the factors in column 13 or divided by those in column 14. The sum of these factored quantities will be the basic rating (column 2) from which the size of water cylinder is selected.

Under conditions requiring lift points in the return; or where there is inleakage around inlet valves or elsewhere; or where large volumes of high temperature returns enter near the pump; or if the run of piping from source of steam supply to farthest radiator is long; or where the radiator traps leak steam; additional factors must be applied to insure the proper size of pump. These factors cannot be summarized since their selection is entirely a matter of judgment and of experience with similar conditions.

Column 4 shows the minimum size for return main entering pump. These sizes are based upon a grade in the piping of 1 ft. in 300 toward the pump and upon a condition where the return pipe is half-full of water and will then discharge condensation by gravity at rates not less than the basic rates in column 2. For these calculations, the Chézy formula $Q = a\,c\sqrt{r\,s}$ is used, in which Q is the quantity discharged, a is the cross-sectional area of the pipe, r is the hydraulic radius, s is the hydraulic slope of the pipe and c is a coefficient.

The size of returns inlet from column 4 will also determine the size of suction strainer which is to be used in this main at the pump.

Column 5 is calculated from the same formula to determine the minimum size of pump discharge and delivery pipe from pump to air-separating tank. In this case the pipe is considered to be half-full of water and its grade is 1 in. in 20 ft.

For purposes of determining the proper size of air separating tank to apply for a given rate of condensation discharge from pump, the assumption is made from field experiences that 1 sq. ft. of liberating surface should be provided under average conditions for each 2100 lb. of water discharged per hour. Column 6 of Table 1 shows the number of square feet of liberating surface required for the basic discharge ratings in column 2.

Where the tank is used only for air-separating purposes such as plain tanks and hydro-pneumatic tanks, the sizes of tanks may be designed directly from the figures in column 6. Dimensions of tanks following this design are shown in columns 7 and 8.

In cases where the tank is used for storage of returns, the tank should be larger than that required for purposes of air-separation only. Columns 9 and 10 show dimensions of such tanks based upon storing the quantities of water which will be discharged during five minutes at the basic hourly rates shown in column 2.

As an example of the complete calculations for sizing the water end of a vacuum pump and for selecting size of auxiliary equipment, assume a group of three buildings, A, B and C, from which condensation flows at rates of 7500, 5000 and 3000 lb. per hr. respectively.

Also assume that the 7500 lb. per hr. of condensation in building A is from blast coils and a closed heater; that the 5000 lb. per hr. from building B is from pipe coils, each containing 130 sq. ft.; that the 3500 lb. per hr. from building C is from direct radiators in 50 sq. ft. units; that return mains are exposed; and that it is proposed to use a plain type of air separating tank.

These condensation rates must be transformed to those which would be realized under the standard conditions upon which this table is based, by means of the factors in column 13, using 0.66 for blower stacks and closed heater, 0.70 for coils larger than 120 sq. ft. and 0.84 for radiator units of 50 sq. ft. By applying these factors the equivalent condensation rates are found to be 4950, 3500 and 2940 lb. per hr. for building A, B and C respectively or 11,390 lb. per hr. for the transformed equivalent total rate.

From column 2, the nearest basic rate is 10,350 lb. per hr. and from column 1, the corresponding diameter of bore for this pump is 10-in.

By dividing the required rate 11,390 by the basic rate 10,350, the capacity factor is found to be 1.10. Going into the table it is found that 1.10 in column 12 corresponds with a relation of stroke to bore of 1.25 (column 11). Multiplying 1.25 by 10 in. (the bore) gives 12.5 in. as the required stroke. The nearest stock size is 12-in. stroke so that a pump having 10-in. bore by 12-in. stroke is selected.

From columns 4 and 5, the minimum requirements for size of returns inlet and discharge for this pump are found to be 4 in. and 2½ in. respectively. If the return main is long, it is better to select 5-in. as the minimm size of return inlet, since 4½-in. is not a regular stock size for pipe and fittings. The suction strainer will be the same size as the return main entering the pump.

Selecting from columns 7 and 8, the size of plain air-separating tank is 18-in. diameter by 48-in. length.

Proportioning of Steam Ends of Reciprocating Vacuum Pumps: In proportioning the steam cylinder, the following is a safe rule to use. *The area of the steam cylinder in square inches times one-third the boiler pressure should equal the water piston area in square inches, multiplied by the combined pressure on the water end (vacuum plus discharge pressure) expressed in pounds per square inch.* This is given by the following equation:

$$A_s \times \frac{P_b}{3} = A_w \times \left(\frac{V}{2} + P_d\right)$$

From which we have

$$A_s = \frac{A_w \times \left(\frac{V}{2} + P_d\right) \times 3}{P_b} \qquad \textit{(Formula 13-1)}$$

in which

A_s = area of steam piston in square inches.
A_w = area of water piston in square inches.
P_b = boiler pressure in pounds per square inch.
P_d = discharge pressure in pounds per square inch.
V = vacuum at pump expressed in inches of mercury.
$\frac{V}{2}$ = approximate vacuum in pounds per square inch (2 in. mercury = approximately 1 lb. per sq. in.)

Note: All pressures are by gauge.

In the above formula, the working pressure is taken as one-third of the boiler pressure, in order to allow for the low mechanical efficiency of the pump, as well as for the inevitable drop in steam pressure between the boiler and the inlet of the pump. Carelessness in setting up the packing in the water-and-air piston is prevalent and to be expected. It is also necessary for the pump to keep going even when the boiler pressure may be considerably lower than the normal working pressure.

While in some cases this formula may give dimensions which appear to be larger than necessary, it is seldom safe to make the area of the steam cylinder less than twice the area of the water-and-air cylinder.

Column 3 of Table 13-1 shows sizes of steam supply and of vacuum governor, for boiler pressure of 75 to 125 lb. per sq. in.

Power-driven Reciprocating Vacuum Pumps: Lack of available steam pressure to operate the piston in reciprocating vacuum pumps requires that some other source of power must occasionally be utilized. Where this is the case, a reciprocating pump is in many cases unsuitable because of the difficulty in handling the varying load during each stroke and because no satisfactory means for controlling the displacement to maintain the desired degree of vacuum has yet been devised for this type of pump.

To move the reciprocating piston in the water cylinder by means of a connecting rod and crank, the latter necessarily rotating at low speed, entails gearing or an extremely large pulley and countershafting. Inasmuch as the torque varies from almost nothing at the ends of the stroke to a high maximum at about three-fourths stroke, back-lash, noise and wear of gears

or slapping and slip of belts are to be expected unless a heavy fly-wheel is used, and in any instance the power consumption is excessive.

Variable-speed motors are sometimes utilized for driving, but are expensive, and give only two or three steps of displacement, which must be selected either manually or by complicated delicate electrical controllers.

There is nothing to commend in intermittent control. Constant speed and displacement with a vacuum breaker to admit air when the load is below normal is probably nearest to a satisfactory arrangement where power-driven reciprocating vacuum pumps are used.

Disposal of Vacuum Pump Discharge: Conditions vary to such an extent that good judgment is the only safe guide in determining the best method for the disposal of the vacuum pump discharge. In no case should the head against the discharge of reciprocating pumps exceed 15 lb. unless the pump stroke materially exceeds the bore and thus reduces the bad effect of clearance. Usually one of these seven methods will best apply:

1. *Discharge to Waste:* Disposal by discharge to waste involves loss of all the valuable heat and water, but in rare cases this is permissible.

2. *Discharge through Air-separating Tanks:* Where first thought seems to suggest disposal to waste, it will in many cases be found possible to deliver the water and air into a separating tank, or stand pipe sufficiently elevated for the water, after separation, to flow by gravity to some point of

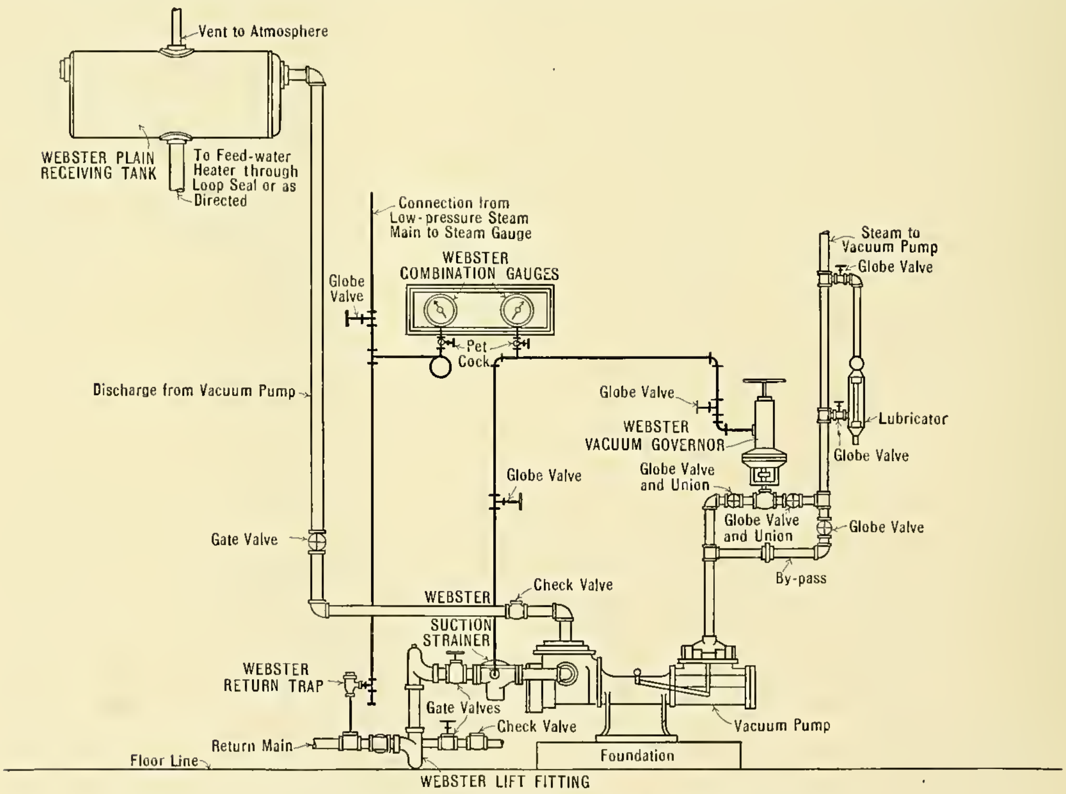


Fig. 13-2. Method of connecting vacuum pump to a plain receiving tank

valuable use, such as boiler or feed-water heater, etc., or for hot water supply.

Where, due to structural conditions, a suitable elevated location cannot be found, the effect of head may be obtained by use of a hydro-pneumatic tank as described under heading No. 4.

3. *Discharge to Open Vent Tanks:* Open vent tanks, otherwise called plain separating tanks, normally serve the purpose of releasing the entrained air from the discharge of the vacuum pump. (See Figure 13-2.)

This air removal requires the generous water surface area of either a tank of large horizontal cross-section, rather than one of large vertical sectional area, or a tank with a large vertical head and enough sectional area to permit of low-velocity downward water flow while entrained air is floating to the surface against the water current, as in a stand pipe. For removal of air, one square foot of horizontal cross-section has usually been found sufficient for each 2100 lb. of water per hour. A stand pipe, with diameter equal to that of the pump cylinder, is usually sufficient, although a more logical rule is to make the cross-sectional area of the stand pipe

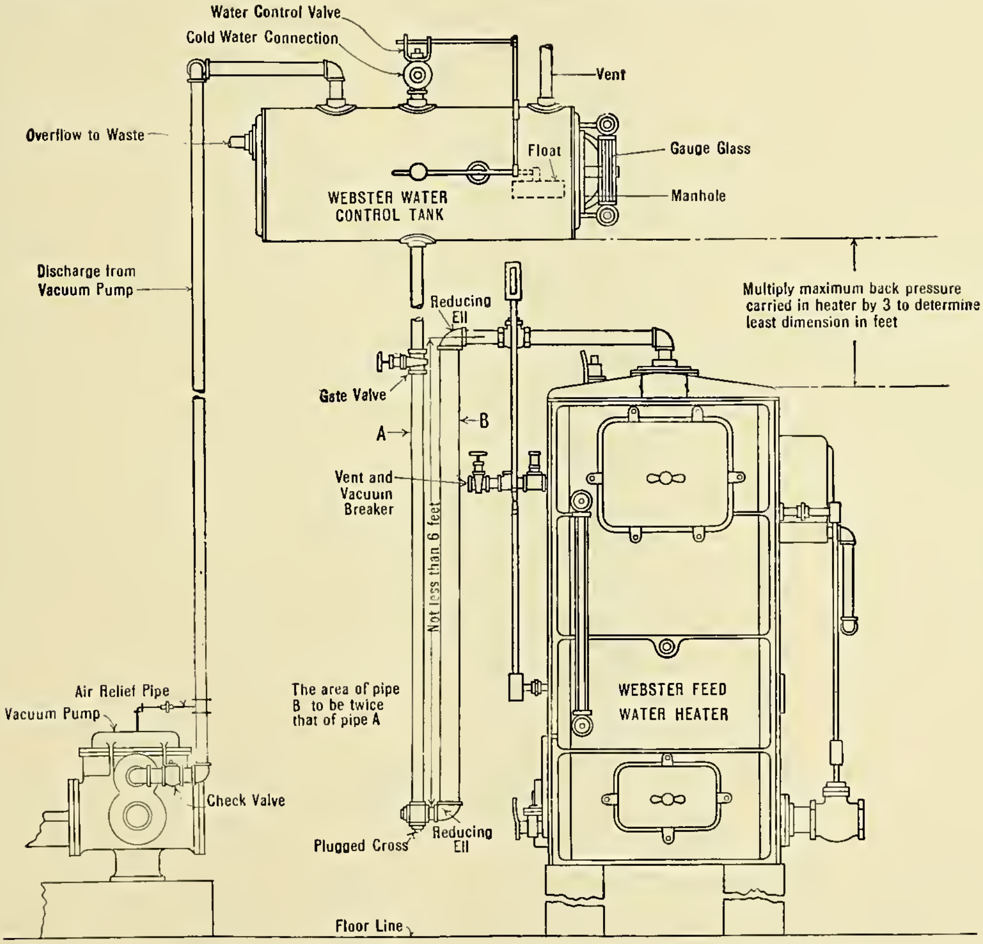


Fig. 13-3. Typical application of Webster Water-control Receiving Tank in connection with an open feed-water heater. The heater should be set on a foundation of sufficient height (a vertical rise of not less than three feet) between the pump outlet of the heater and the suction valves of the boiler-feed pump

bear some direct relation to the amount of condensation from which the air is to be separated, and to the height of column of water through which the air bubbles must rise against the flow of liquid.

The fact that the discharge of reciprocating wet-vacuum pumps is a mixture of water and air favors the use of a freely vented separating tank wherever a suitable location may be obtained. This is such height that the pressure produced by the water column will be sufficient to overcome that in the low-pressure boiler, feed-water heater (see Figure 13-3), or other point of disposition.

The effective column or head between the pump-discharge valve and the inlet of the separating tank will be less than that of solid water by the volume of air contained in the mixture. The contents in separating tank and discharge pipe therefrom will be water only. It is, therefore, possible with pump discharge properly proportioned and provided with lift fittings, vertical rise pipe to tank, etc., to obtain a gravity head in the tank discharge above the level of the pump valve deck, considerably greater than the pres-

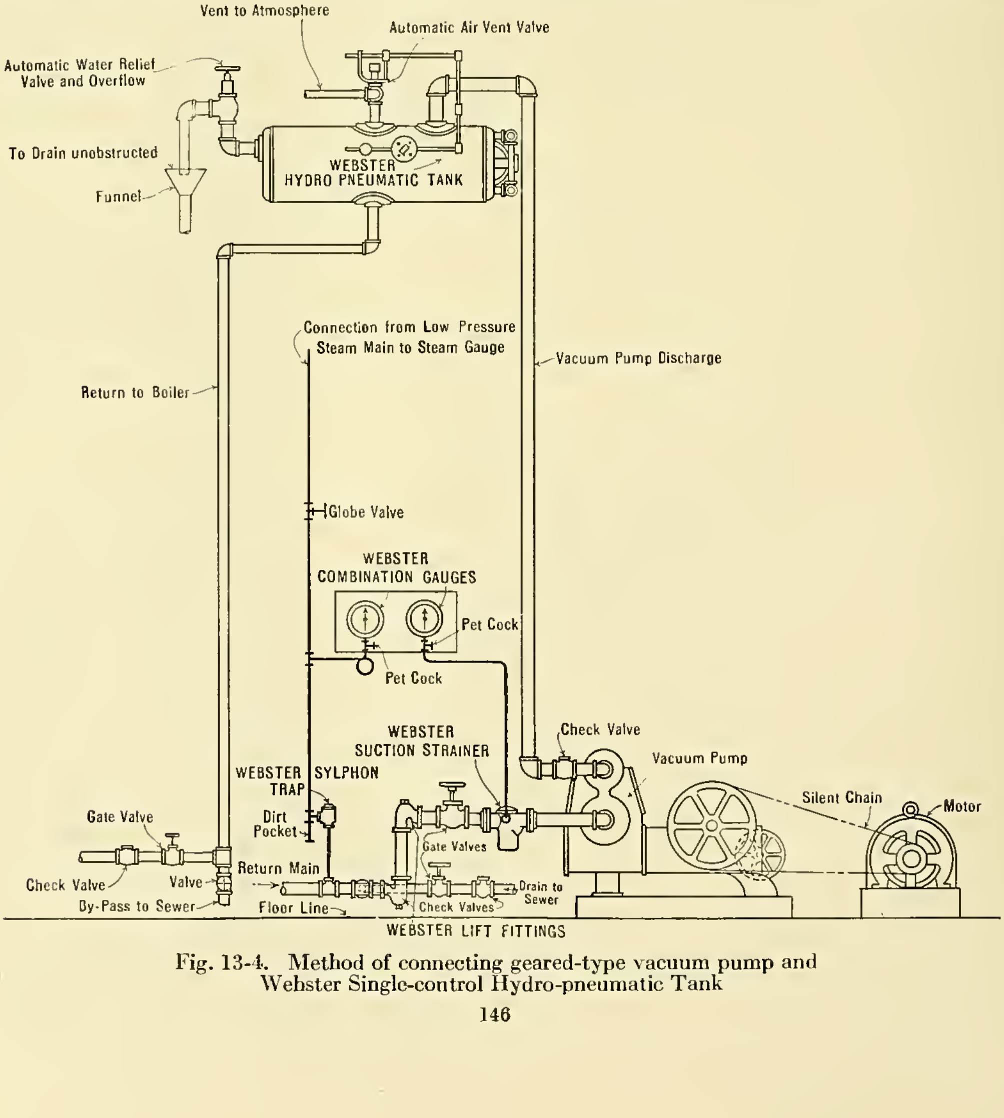


Fig. 13-4. Method of connecting geared-type vacuum pump and Webster Single-control Hydro-pneumatic Tank

sure in the pump cylinder necessary to lift the valves and discharge the condensation to the elevated return tank.

4. *Discharge to Hydro-pneumatic Tanks:* As the name indicates, hydro-pneumatic tanks bring the elastic pressure of the liberated air to act on and supplement the head, in the discharge of the water of condensation. A float-controlled valve is placed on the air outlet of the separating tank, and so arranged that when the water of condensation has not sufficient head to flow by gravity to the point of use, the air will be confined in upper part of tank. As the pump continues to deliver water and air to the tank (see Figure 13-4) the pressure inside the tank increases until sufficient to discharge the water, thus lowering the water line and eventually permitting escape of the surplus air through the float-controlled air valve.

The discharge of condensation to low-pressure boilers, in which the pressure may at times be less than that of the atmosphere, requires another float in the hydro-pneumatic tank (see Figure 13-5) to control the valve on the tank water discharge and keep this pipe closed at such times as there might be danger of air flowing from the tank to the boiler.

The hydro-pneumatic type of tank is used only where an open tank cannot be located at a height sufficient to provide gravity head to discharge the tank contents against the maximum pressure in the heater or boiler, or

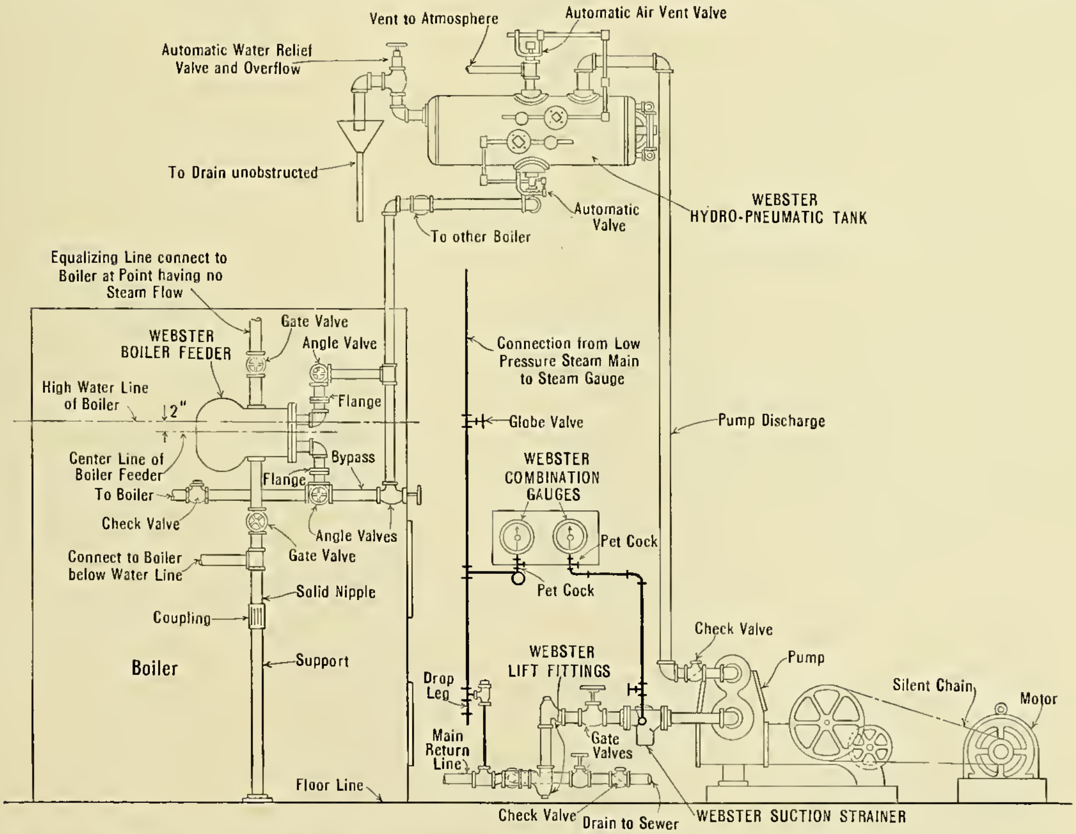


Fig. 13-5. Typical connections to vacuum pump, double-control hydro-pneumatic tank and boiler feeder

where there are large variations between the maximum and minimum pressures to be overcome. Where the hydro-pneumatic tank is used merely as a substitute for an open separating tank, little advantage may be taken of the light density of the pump discharge.

The confined air pressure in the hydro-pneumatic tank plus the gravity head in the tank discharge pipe must be sufficient to cause flow to the place of disposition. This confined air pressure plus the column of mixed air and water in the pump discharge to the tank is the total head against which the pump must act.

Where pressure on the heater, boiler, etc., varies materially from time to time, but in general is near the minimum, a substantial saving in energy may be obtained by using a hydro-pneumatic tank instead of a plain tank set at higher elevation to overcome the peak pressure in the boiler or heater. The use of a plain tank under these conditions keeps the pump operating constantly against the maximum head, where a hydro-pneumatic tank set lower operates as a plain tank whenever the gravity head in the tank is sufficient to cause flow at the low elevation, and employs the combination of air pressure and gravity head (with air vent closed) only at times of peak load. Only then is the air pressure load added to the pump discharge.

5. *Discharge to Loop Seal on Tank Outlet to Heater or Boiler:* The disposal of water of condensation from a return tank to a feed heater (see Figure 13-3), boiler or other receptacle, in which there may be greater pressure than that of the atmosphere, requires guarding against back flow of steam, air or whatever other elastic fluid may be present at the outlet.

A loop seal has been found most suitable for this purpose, provided the seal is made long and contains ample volume in the vertical leg on the pressure side. A variable pressure when increasing tends to force the level of water down in the leg on the pressure side and up in the leg toward the tank. If there is not sufficient water in the loop, the water will become displaced, and the seal broken before enough of a water column has been built up in the leg from the tank. The column will then blow into the return tank and the steam or other elastic fluid will continue to blow while its pressure is above that at the tank outlet.

The fact that water in the tank is ready to seal the loop below will not avail as long as there is a difference in pressure between the tank and boiler sufficient to blow a comparatively short slug of water back into tank. The only way to restore the seal is first to equalize the pressure on both legs. A good practice is to proportion the leg on the pressure side to hold twice the contents of the pipe from the tank to the bottom of the seal.

6. *Discharge to Receiver and Boiler-feed or Tank Pump:* Where the head on the delivery side of steam-driven vacuum pumps exceeds 15 lb., it is good practice to deliver the condensation to a vented receiver (see Figure 13-6) located close to the level of the vacuum-pump outlet. This receiver should be connected to a separate steam or power-driven water pump which is capable of delivering against the maximum head. (See Figure 13-7.) If this pump is steam-driven, its displacement should be controlled by a throttle valve, actuated by the water line in the receiving tank; if power-driven, the effective displacement may best be controlled by bypass valve between

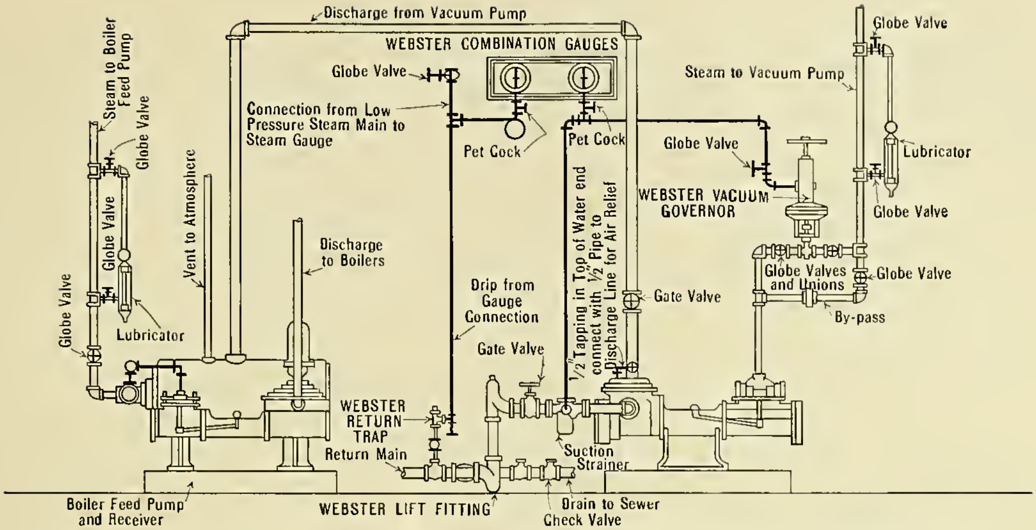


Fig. 13-6. Method of connecting vacuum pump and automatic boiler-feed pump and receiver

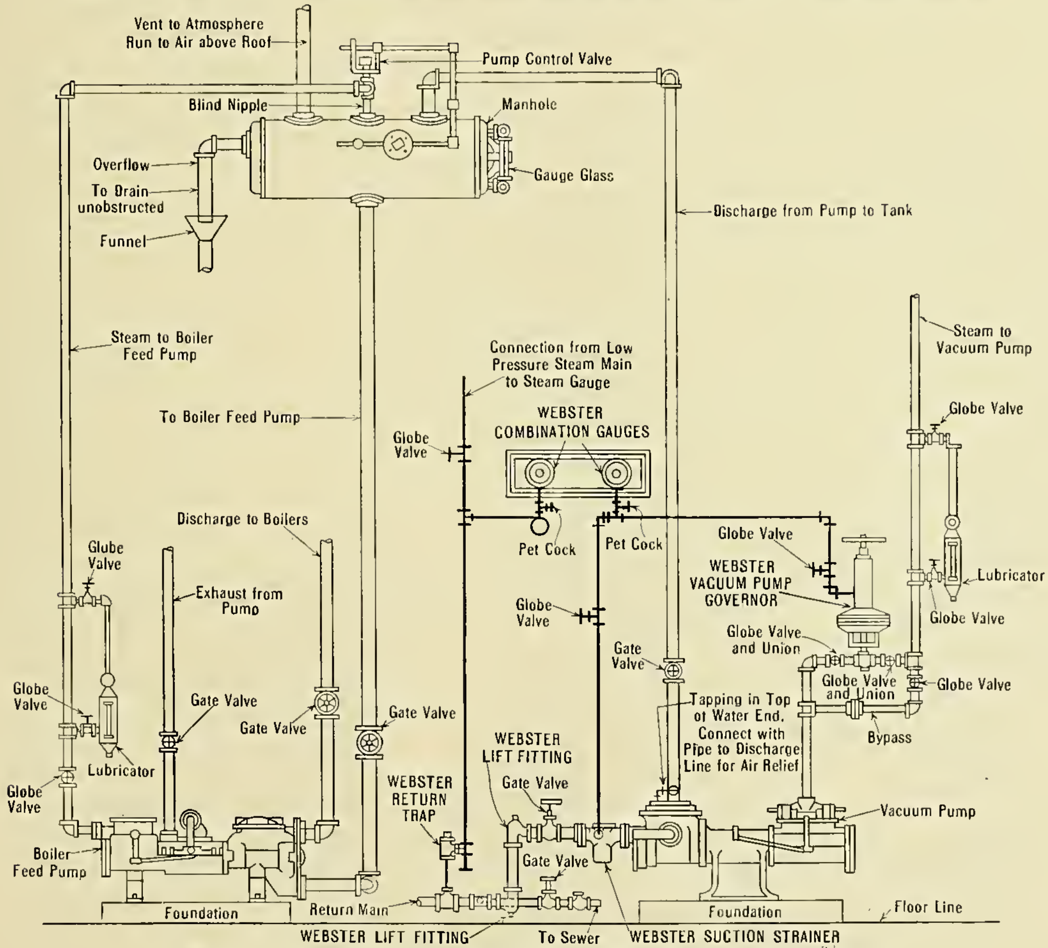


Fig. 13-7. Method of connecting vacuum pump, boiler-feed pump and Webster Steam-control Receiving Tank

pump suction and delivery, and actuated by water-line float in the receiver.

7. *Dry-vacuum Pump Receiver and Water Pump:* This combination proves very effective under conditions of high delivery head where the main

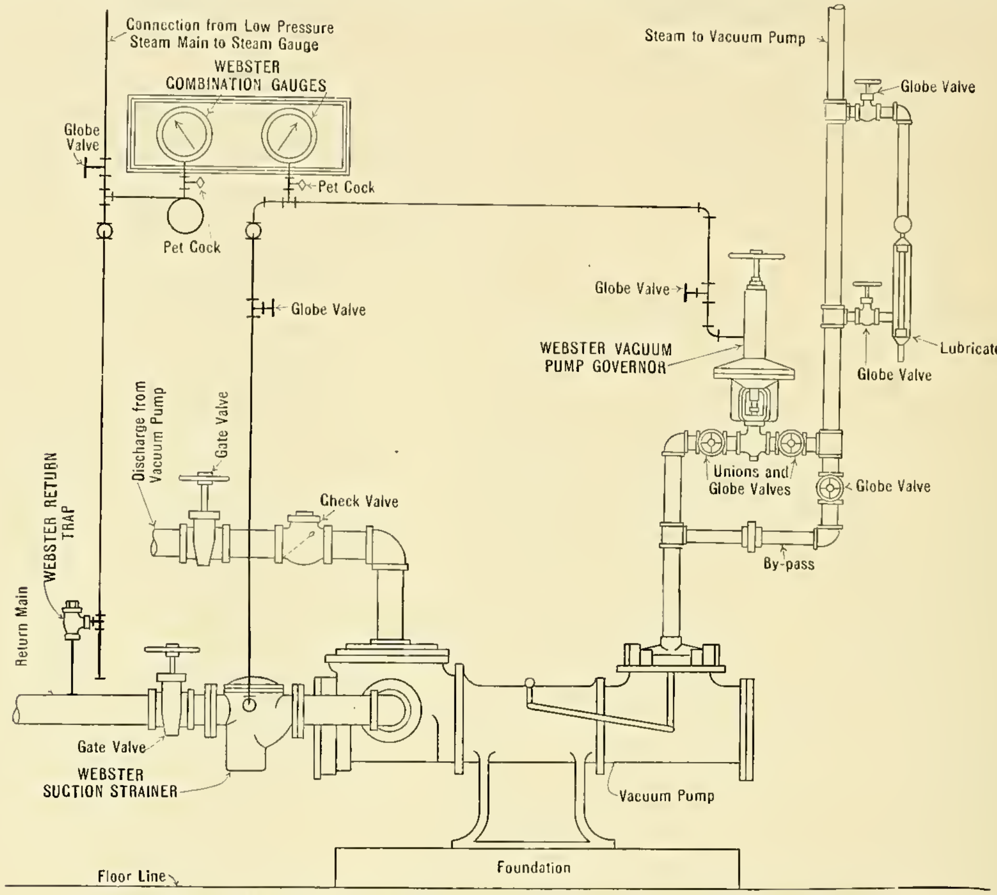


Fig. 13-8. Method of making connections to steam-operated vacuum pump

return can be arranged to flow by gravity to a closed receiver, which in turn is sufficiently elevated above the location of water pump to provide a head of 2 to 3 lb. on the pump inlet valves.

The dry-vacuum pump being free from dirt and abrasive material, may have close clearance and fairly high efficiency. It may be located above and take its suction from the top of the receiver, and frequently some form of condenser may be arranged in the suction line to absorb and utilize otherwise wasted heat from the air and water vapor and at same time materially reduce the volume of vapor to be handled.

The receiver, if properly designed, forms a receptacle for the grit and impurities which would otherwise injure the water pump; and it also affords space for a float governor for controlling the water pump by the varying volume of return water.

Excessive vacuum in the receiver will cause trouble in the water pump. For this reason, a vacuum governor should always be used to control the dry-vacuum pump and to hold the vacuum within pre-determined limits.

SUCTION STRAINERS: The worst of the grit and dirt from condensation should be retarded and removed before entering the pump where it would score the water cylinder. Strainers (see Figure 13-8) with readily removed baskets for use on the main vacuum return line were first designed and recommended by Warren Webster & Company 24 years ago. The original Webster design with little modification has been almost universally adopted.

In some instances, conditions arise where large quantities of returns, at unusually high temperatures, are discharged into the line near the vacuum pump. These may come from special apparatus such as cooking or hospital fixtures, dry kilns, or other devices using high pressure steam. A combination of suction strainer and a cooking device, shown on page 262, will be found to be of advantage, particularly where it is desired to carry a high vacuum at the pump. Cold water, passing through copper coils, is used to condense the vapor in the main return.

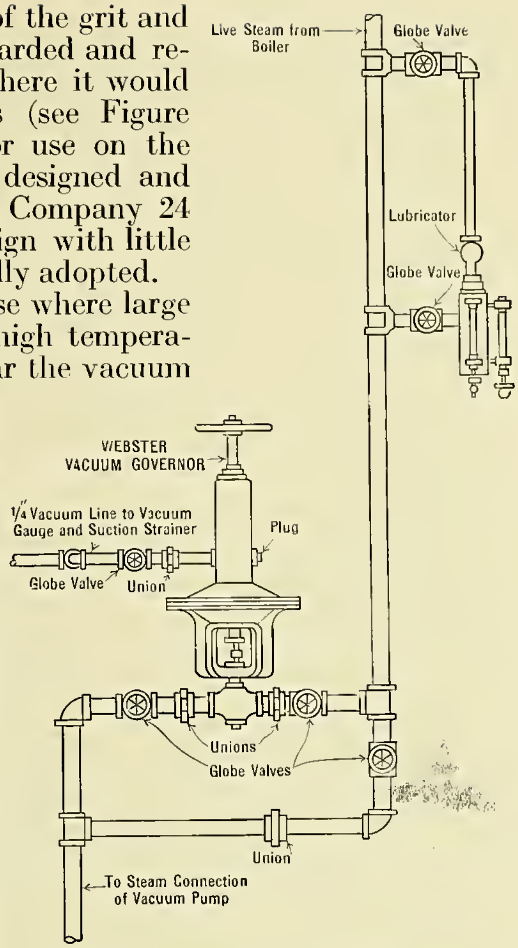


Fig. 13-9. Connections for a Webster Vacuum-pump Governor

VACUUM GOVERNORS: In steam-driven pumps, control of displacement by the degree of vacuum maintained in the return line may be effectually accomplished by throttling the steam supply. (See Figure 13-9.) Simple forms of diaphragm-actuated throttle valves will control the degree of vacuum in the main return within sufficiently narrow limits for all practical purposes.

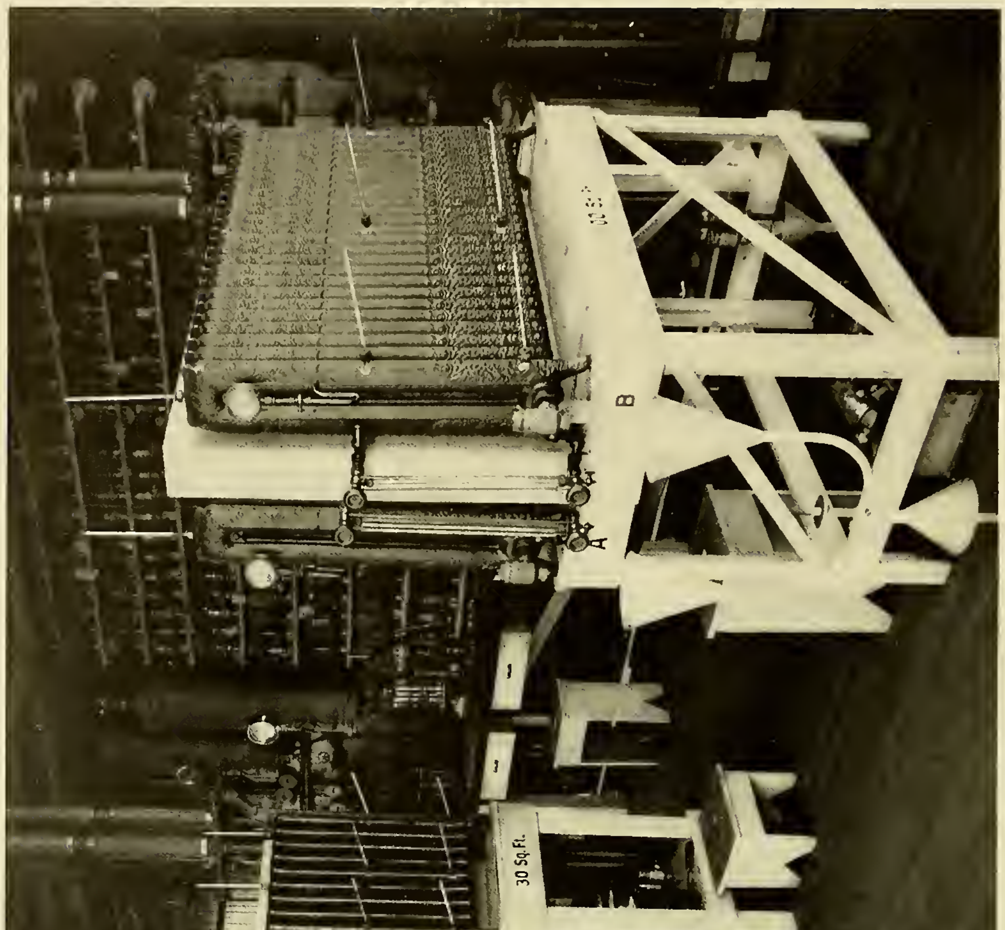


Fig. 14-1. Apparatus for testing in the Mechanical Laboratory of Warren Webster & Co.

CHAPTER XIV

Laboratory Tests of Return Traps

THE object of laboratory tests of appliances is to determine the efficiency of the apparatus tested, as a guide to judgment in selecting materials or in the case of technical schools, as a part of the instruction of the students in methods of scientific research.

All of the operating conditions possible or probable in an actual heating system cannot be artificially produced in the laboratory, nor is it practical to carry out tests long enough or upon sufficient numbers of samples to learn all facts which become evident in practice. Furthermore, as the whole heating system, including design and installation, has its effect upon the efficiency of the devices entering into it as parts, any laboratory tests for efficiency can indicate only the results which are probable when the devices are properly used in practice.

Too much stress should not be laid, therefore, upon the comparative performances of any two makes of traps during laboratory tests. Knowledge of performances in actual installations of many heating systems, maker's ability and care in manufacturing, shop tests, inspection and proper engineering application of the traps are of great importance to the investigator who wishes to make commercial use of his study of such devices.

However, as laboratory tests have their useful place in commercial investigation, the various types of traps and the results of tests which may be expected are outlined in this chapter. Mention is made of many common forms of tests which give erroneous results so that these errors may be avoided. Methods and apparatus for reliable tests are mentioned and illustrated.

Usually the object of a laboratory test of a return trap is to determine one or all of the following characteristics:

1. Effect of the trap upon radiator efficiency.

2. Efficiency of the trap for the removal of air and water of condensation and for conservation of steam and vapor.

3. Behavior of the trap without special adjustment to meet the varying conditions of pressure and vacuum in normal practice.

4. Durability of the trap through a long period of use.

5. Construction features of the trap, particularly the amount of valve movement, which indicates the ability to get rid of dirt and pipe scale.

The results of tests by many investigators, of radiator and trap efficiency, have varied widely and have often been misleading, largely because the methods of testing have been faulty and partly because the devices themselves have not always been manufactured to operate uniformly.

Most tests of which the results have been published have been faulty through failure to cover a wide enough variety of test conditions, through limitation of the time period for each test to a few minutes instead of hours, and through considering and testing only one or two samples of any one

device, instead of six or more selected by the investigator from the manufacturer's stock bins or purchased in the open market.

TESTS FOR HEATING EFFICIENCY: The heating efficiency of a radiator depends upon physical conditions within the radiator which are affected by the action of the return trap. The radiator, among a number of common size and type, which maintains the highest average temperature when tested under the same conditions, is the most efficient.

The greatest possible steam economy is obtained where this efficiency is highest; that is, where steam is being condensed to the greatest extent possible within the radiator and the trap passes the least amount of steam or vapor into the return pipe.

The highest radiator efficiency can be obtained only where the discharge is sufficiently and properly restricted to prevent steam from blowing into the return. Also the air released from the steam in the radiator must be allowed to settle to the lower parts, from which it can enter the trap and be discharged.

A return trap, in addition to restricting the discharge, must effectively accomplish the following:

1. The discharge of all water of condensation as formed. Otherwise water accumulates in the radiator, prevents free discharge of air and thus reduces the amount of surface effective for emitting heat from the steam.

2. The discharge of all air and other gases from the radiator immediately upon their reaching the discharge outlet.

3. Thorough prevention of the discharge of steam to the return.

To accomplish these requirements the valve of a return trap must open or close within a very narrow range of temperature, above or below that of steam at pressure, irrespective of variations in steam pressure, and must adapt itself to such changes of pressure and corresponding steam temperature as may be met in practice.

A brief review of the various types of return traps will facilitate a better understanding of tests and the results which are desired.

All return traps commonly used in low-pressure or vacuum steam heating practice may be classed as float, differential, and thermostatic traps.

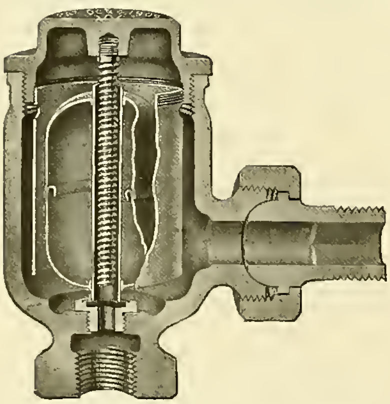


Fig.14-2 Float trap with sealed float

Float traps may have sealed floats, Figure 14-2, or inverted open buckets as the means of operation. In either case, the float is raised by incoming condensation to uncover the valve seat through which water is discharged. Air escapes into the return pipe through an air port, which must be located above the highest water level in the trap. The air port is controlled in some makes by thermostatic devices to prevent leakage of steam to the return.

Tests upon a float trap may generally be expected to show considerable leakage of steam to the return unless the air port is thermostatically controlled. If the air port is so

controlled, the small port and its mechanism may be vulnerable to the effects of dirt and rust. Such traps, however, will be found to have large water discharge capacities and some of the various makes can be used to advantage where widely varying volumes of water must be discharged without respect to temperature.

A differential trap depends for operation upon the difference in pressure at the inlet and at the outlet. In its simplest form, it is a check valve which is closed when the difference in the pressures ahead and back of the clapper is insufficient to overcome the weight of the clapper, Inasmuch as no special means are provided for discharge of air, such a valve may be expected to leak steam to the return under any conditions of higher differential pressure, and to stay closed with consequent air binding and water logging of the radiation when the pressure differential falls below the predetermined limit for which the valve is adjusted.

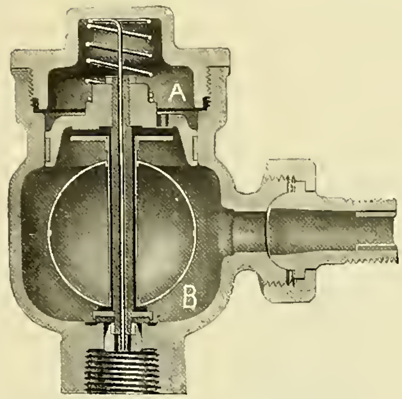


Fig. 14-3. Differential trap with float and piston

Another form of differential trap is shown in Figure 14-3. Water entering the valve body raises the float, thus closing the air port by means of the valve piece attached to it. A higher pressure in the lower part of the trap B than that existing in the chamber A results in the operation of the piston which raises the valve from its seat by means of the connecting valve stem. As the condensation is discharged, the water level lowers and causes the float to fall, thus uncovering the air port, and equalizing the pressures on opposite sides of the piston. The weight of the operating parts and the force of the spring then closes the valve. This trap may be expected to show fairly good results in laboratory tests but it is not satisfactory under the usual operating conditions in which dirt and scale are always present.

A thermostatic trap depends for its operation upon the difference between the temperature of steam at the pressure in radiator, and the temperature of the condensate or air to which the thermostatic member is exposed.

Many devices have been made which depend upon the expansion and contraction of metals or composition, or which make use of a bourdon tube. As a class these have failed because there is not enough difference in area between the inside and outside of the spring to produce the required force at normal difference in temperature between steam and air vapor at a given exterior pressure. This and other faults, such as the necessity for adjustment for varying pressure conditions and slowness in operation, have led to the abandonment of these types by most manufacturers.

Of all types of return traps, the ones in general use today are those which depend for movement of the valve piece upon the change of vapor pressure of fluids confined within a flexible chamber when subjected to different exterior pressures and temperatures. The volatile fluids contained in the flexible chamber vaporize to a greater or less pressure depending upon the

temperature of the steam, vapor, water or air which surround the chamber. The expansion or contraction of the chamber moves the valve piece which is attached to the free end of the chamber.

These traps are, generally speaking, of either the "inboard" type where the thermostatic member is exposed to the temperature and pressure of the steam, water and air as it exists at the radiator outlet, or of the "outboard" type which depends for operation upon the conditions existing between the valve piece and the entrance to the return piping beyond the trap.

To be effective for the inboard type, the thermostatic member must expand and contract through a distance sufficient to open and close the valve under the influence of the extremely small differences of temperature which exist during normal operation. Most traps of the inboard type are inefficient because of the very short "stroke" which can be realized with the inelastic disc construction generally utilized for the flexible chamber, this defect resulting in inability of the trap to rid itself of dirt and scale.

Traps of the outboard type are affected by the pressure and temperature of the return. They are in proper adjustment only at one definite pressure and temperature and out of adjustment at all other normal combinations of pressure and temperature. They cannot be adjusted even for these normal variations in radiator pressures and vacuum in the return, and as a result usually water-log and air-bind the radiator by staying closed when high temperature and pressure exist, or stay open and blow steam under conditions of low temperature and pressure.

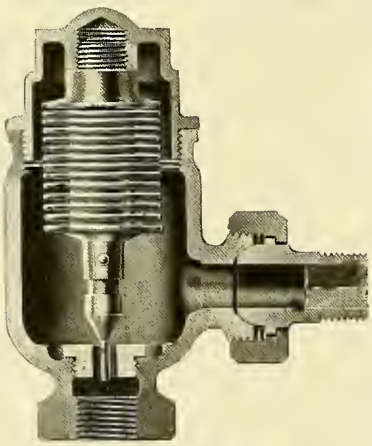


Fig. 14-4. The Webster Sylphon Trap

The trap shown in Figure 14-4 is a thermostatic trap of the inboard type and as such is affected in operation only by the temperature and pressures existing within the radiator. The multifold design of the thermostatic member gives it great elasticity and consequent ample movement in response to change of temperature and pressure in the medium surrounding it. This member contains liquid which makes the trap self-compensating for difference in operating pressures of steam within the radiator. Its construction, with conical valve piece seating on sharp-edged seat, assures positive self-cleaning. Dirt and scale cannot lodge between valve and seat and permit steam to leak into the return.

It has been stated that a trap must not leak steam to the return, but in this connection there should be no confusion between steam discharged through a trap and vapor rising from hot condensate. Though their appearance during certain forms of visual tests are much alike, they are two entirely different things, and if confused with each other, as is sometimes done, wrong conclusions will result.

Many times, highly efficient radiator traps are condemned for leaking steam, due to the observed vapor of re-evaporation noted at their discharge outlet, and less efficient traps have been commended because of absence of such vapors.

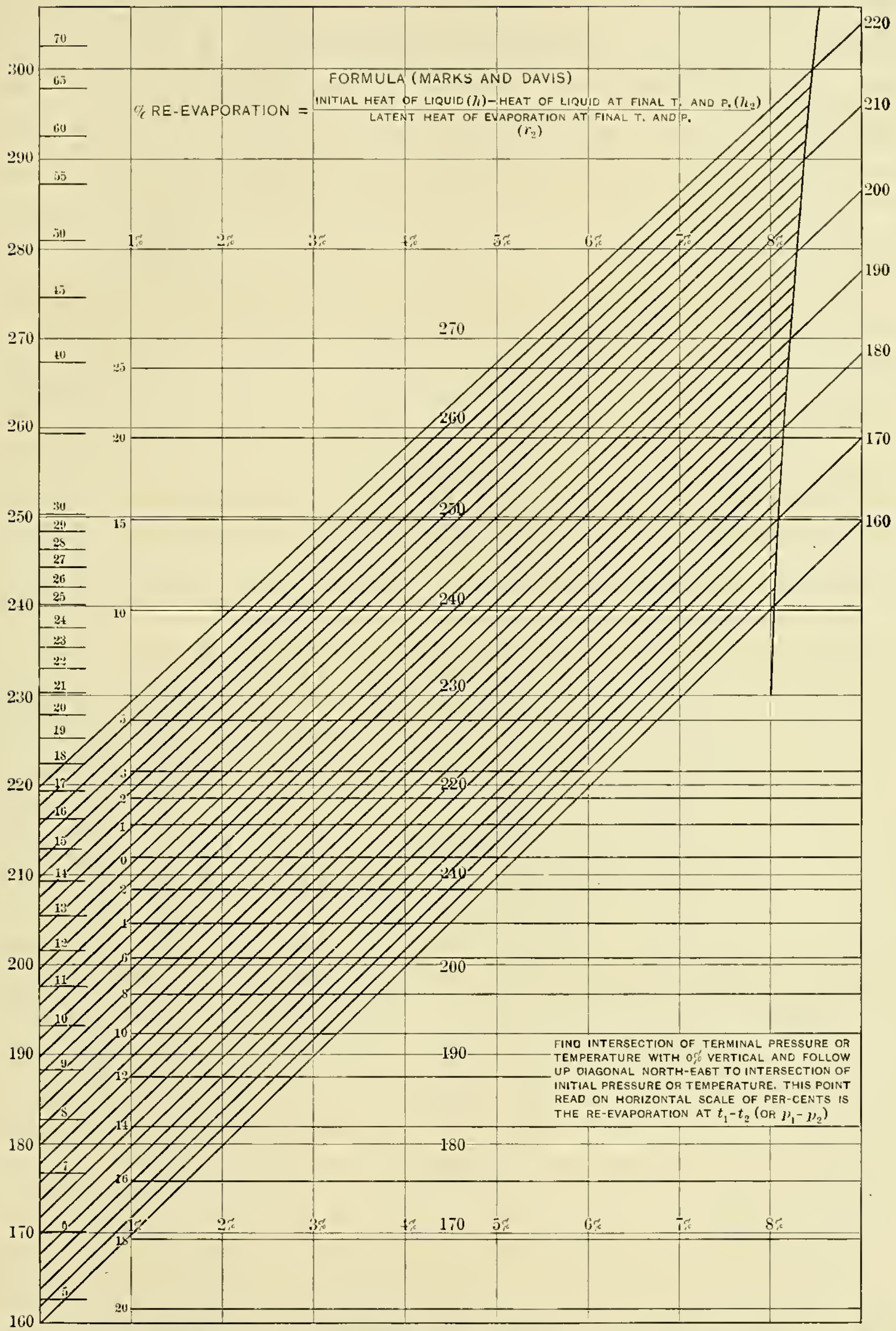


Fig. 14-5. Re-evaporation chart for determining the percentage of water re-evaporated from any temperature between 300 and 170 deg. fahr. into water vapor of a lower temperature and corresponding pressure

The absence of vapor at the discharge is in reality an indication that the trap is holding back condensation and entrained air until the temperature of the discharge is materially less than that of steam at the pressure of the outlet. The consequence of such holding back is a partially air-bound and water-logged radiator, with less than full radiating efficiency.

Visibility is deceptive. A great amount of moisture in the atmosphere and favorable light conditions both add to the visibility. The air discharged from an efficient trap is saturated with water at discharge temperature and this water mixing with air at room temperature looks like steam, while the discharge of a trap utterly deficient in air removal shows only the vapor of re-evaporation.

The water of condensation contains total heat in excess of that in water of condensation at lower pressure. This excess heat boils off some of the condensation into steam. The amount so boiled off is entirely dependent on excess of total heat in outflowing condensate above total heat of water at lower pressure.

If steam passes out with condensate, a steam of greater total heat is dissipated. A fully efficient trap releases the condensation at or near steam temperature and radiator pressure, into a return of lower pressure. All heat above that consistent with lower pressure then generates vapor. This vapor passes to the vapor receiver in a test. A certain amount of vapor per pound of condensation is normal and any excess of vapor above the normal is steam leakage.

The condensate from a higher pressure into a lower pressure will never be at a higher temperature than that due to steam at the lower pressure. The excess of the heat in the outflowing condensate will flash part of the water into steam.

These points are emphasized to show the fallibility of visibility test to show the efficiency of return traps.

Very rough tests are often made by connecting a trap to the end of a pipe or to outlets in a header to which steam is admitted at the pressure usually used, the trap discharging into the atmosphere. A test of this kind merely shows whether the trap shuts off.

Comparative values are sometimes placed upon traps by considering the quantity of water discharged during equal periods of time. The traps are successively attached to the same test radiator, the condensate is carefully weighed and the conclusion drawn that the trap passing the largest quantity in a given time is the best. It is evident that such a test shows merely the condensing rate of the radiator under the room temperature conditions. Nothing is demonstrated regarding the performance of the trap, for it is only when condensation is held back in the radiator that the capacity of the trap is exceeded. This test is only a determination of the condensate-discharging capacity of the trap.

The vacuum which can be maintained at the discharge end of a trap is occasionally regarded as a criterion of the comparative worth of traps. For such tests, the apparatus consists of a radiator, a return trap, a return connection to a vacuum pump, and devices for maintaining constant pressure of steam supply to the radiator and for operating the pump at a constant

speed. The trap maintaining the highest vacuum during the test is considered to be the best. With little or no attempt to determine the extent to which the radiator is air and water bound, such data has frequently led to a wrong choice of traps and the results when in actual operation on a heating system have proved correspondingly unsatisfactory.

Another test is to connect a trap to a radiator with discharge to atmosphere, and noting the operation.

Particularly erroneous conclusions will be reached unless careful distinction is made between the vapor which is steam and the vapor which is due to re-evaporation.

Much can be learned as to trap behavior from such a test, yet the conditions are often not the same as in actual service operation. The return piping connection and the pressure therein have considerable effect upon their operation so that rough tests of this nature should not be accepted as conclusive, but as indicative of trap operation.

These few devices and methods are the ones commonly used for determining comparative worth of return traps where only the most easily procurable testing apparatus is available. Like other scientific investigations more careful methods will lead to more reliable results and with proper apparatus and thoughtful procedure it is entirely practicable to obtain test data which can be relied upon as accurately forecasting the success which may be expected from the use of any return trap in an actual heating system.

The first thought for any reliable test should be to create laboratory conditions as nearly as possible like those met in actual practice. Coincidently, the apparatus should be designed to provide exactly like and simultaneous test conditicns where traps are tested for comparison, and of course, appliances for measuring the results must be carefully placed and adjusted. Then, by following a proper test, planned to exhaust the various possibilities of different operating conditions, results are secured which can be accepted as conclusive.

Enough has been said to show that valuable data regarding the probable performance of return traps can be obtained in the laboratory where suitable apparatus is available and where suitable test methods are carefully applied. However, the long-time test of devices in actual heating systems is the best guide for determining the relative value of return traps, and further, the efficiency of a good return trap can be fully realized only when the heating system itself is properly planned and operated.

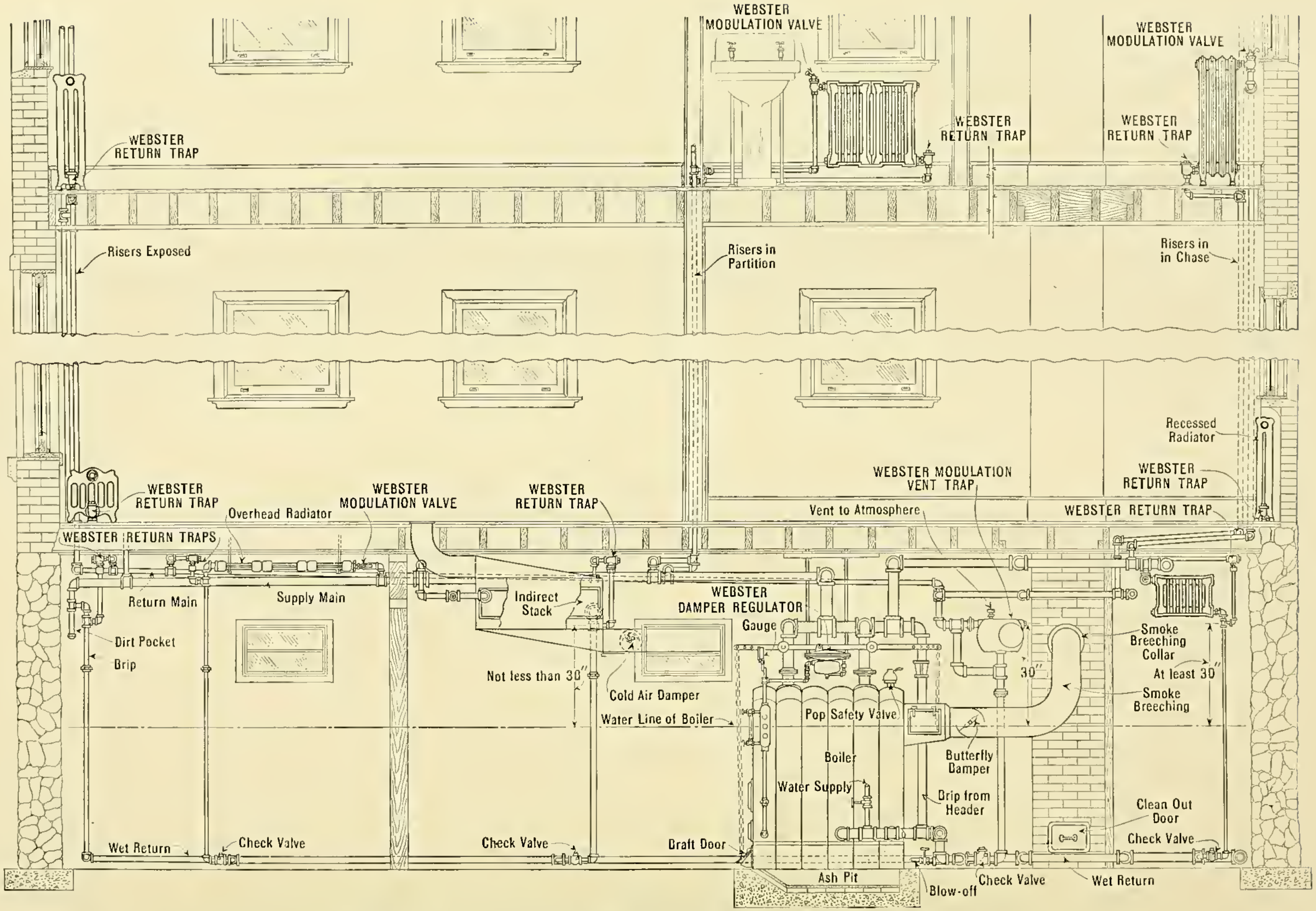


Fig. 15-1. Typical lay-out of a Webster Modulation System for a small building using a cast-iron boiler

Part II. Webster System Specialties and Applications*

CHAPTER XV

Webster Systems of Steam Heating

THE title "Webster Systems of Steam Heating" is used to designate not only the Webster Specialties which are used in the several types of heating systems, but also the methods and arrangements, most of them original with the manufacturer, which assure economical and efficient operation of the heating plant as a whole.

In addition this designation embraces a far-reaching policy of co-operation—Webster Service—which is rendered through branch offices and service centres of the manufacturer in the principal cities.

This three-fold system of specialties, methods and service is the result of continuous development since 1888.

Many of the methods of application have been reduced to the form of Standard Service Details, as shown in Chapter 22 and elsewhere in this book.

The selection and adoption of a Webster System carries with it the assurance to the architect, to the designing engineer, to the heating contractor and to the owner, that the responsibility is not divided between manufacturers of various appliances.

In a Webster System all of the appliances are co-ordinated in their application and function, and the great risk of patchwork selection and responsibility is avoided.

Webster Specialties have been proved by the test of use over many years to be the highest quality attainable in design, workmanship and material.

Webster Service and the standard and special details of recommended application are the result of long experience and pioneering in solving the practical problems that have arisen.

Webster Systems are flexible. There is a type or a modification that will fit each building. Following the classification in Chapter 10, Webster Systems of Steam Heating are divided into two general types: Webster Modulation Systems and Webster Vacuum Systems.

Webster Modulation Systems

As stated in Chapter 10, the vacuum and modulation types of steam heating systems are sufficiently alike to be classed as one broad type of system, in which the circulation of steam is produced by a flow of the heating

*Drawings showing applications, and dimensions of apparatus are subject to change without notice. Certified drawings of apparatus will be furnished upon request.

medium from a higher to a lower pressure. They are dissimilar in the method of disposing of the products of condensation.

The Modulation System may be sub-divided according to source of steam supply, or more particularly type of boiler, into three general classes:

1. Low-pressure heating boilers operating up to 10-lb. pressure.
2. Boilers operating at from 10 to 50-lb. pressure.
3. Street systems, carrying any pressure.

1. Boilers Operating up to 10-lb. Pressure: A typical arrangement of the Webster Modulation System as installed in connection with a low-pressure heating boiler is shown in Figure 15-1. The initial pressure is closely controlled by means of an extremely sensitive Webster Damper Regulator. The steam is admitted to each radiator through a Webster Modulation Valve which permits modulation of room temperature by simple hand manipulation. Condensation is discharged and air is vented from each radiator through a Webster Return Trap which maintains full heating efficiency of the radiator and eliminates the annoyance, difficulties and noises common to ordinary gravity steam heating systems.

Condensation and air from each radiator flow by gravity through a system of return risers and mains into the Webster Modulation Vent Trap, where the air is automatically vented, permitting the system under favorable boiler conditions to operate for long periods under partial vacuum or "vapor," but also due to the flexibility of the system permitting higher pressures to be carried in severe weather when a maximum amount of heat is required. Fig. 24-61, Page 268, shows the detail connections of the Modulation Vent Trap.

The system of supply and return mains and risers should be sized and run as recommended for Modulation Systems in Chapter 11. As a general rule, supply mains and risers are not dripped through traps, but directly into a wet-return line, the air being vented into the dry-return line which is run back above the boiler water line to the Modulation Vent Trap.

Where building conditions make the running of a wet-return line impossible, the mains and supply risers are dripped and vented through Webster Return Traps into the dry-return line. It has however been found preferable from practical experience to run a wet-return line wherever it is physically possible to do so.

In view of the general adoption of Webster Modulation Valves and the hot-water types of radiators, the top feed supply connections are more generally used. When placed in this position, the valves are in a very accessible location and it will be found easier to control the temperature of the room by operating the valve than by following the customary method of opening and closing the window.

Figure 22-43 on page 228 illustrates the method of dripping and venting the supply main into the wet return. Figures 22-45 and 22-49 on pages 229 and 232 show how the basement radiators are connected up to the system. Several methods of dripping the risers and mains through Return Traps into the dry return are shown in Chapter 22 on pages 215, 216 and 217.

The Webster Modulation Vent Trap is essentially a part of the Webster Modulation System, to be used on installations where the sizes of pipes,

valves and return traps have been computed in accordance with the methods explained in Chapter 11 and also on the basis of pressure differential outlined in Chapter 23, and summarized in Table 23-7.

Where the pressure difference between that in the boiler and that in the main return line is likely to exceed the available gravity head between the return main and the boiler, the Webster High-duty Vent Trap may be required.

The principal conditions under which the High-duty Vent Trap may be employed are as follows:

1. Where it is of advantage to design the system for a continuous operating steam pressure ranging from 2 to 3 lb. to occasionally 10 lb.

2. In an existing installation, where the pipe sizes are already fixed, as for example an old building in which complete steam circulation cannot be obtained under 2 or 3 lb.

3. In a proposed installation where the basis upon which the pipe sizes, valves and return traps are figured is either uncertain or unknown.

4. Under certain operating conditions such as continually changing janitor service, operating the boiler without the use of a sensitive low-pressure damper regulator or with the damper regulator entirely detached.

5. In cases where special grades of bituminous coal are burned in certain types of boilers, and it is impossible to maintain low steam pressure even with careful attention and correct damper regulation.

2. Boiler Pressure from 10 to 50 Lb.: With this type of system the heating medium is generally live steam taken directly from the boiler and is reduced to the desired pressure, varying from atmospheric up to 1 or 2 lb., by means of a pressure-reducing valve. This initial pressure in the heating main will vary according to the pressure drop for which the supply piping has been sized, and to a certain extent with respect to the outside temperature and weather conditions.

The only exhaust steam available is that from boiler-feed pumps and other auxiliaries if steam-driven. The exhaust is utilized after it has been made suitable for use by passing through a Webster Oil Separator, drained by a Webster Grease Trap.

The system of supply and return mains and risers should be sized and run as recommended for Case 1.

In small and moderate size buildings the supply mains are usually run on the basement ceiling and connected through laterals to up-feed risers supplying the radiators.

In tall buildings and in buildings of certain types it is desirable to avoid running the large supply mains on the basement ceilings. Where a building is spread over a large area, if the supply main is located on the basement ceiling, the pitch required by the main and by the dry return may cause the latter to be too low when approaching the point of discharge. In both of these cases, what is known as the "overhead" or down-feed system is employed, the steam being fed through a main up-feed riser to a distributing main located at the ceiling of the top story or preferably in the attic, steam being delivered to the various radiators through a series of down-feed risers.

The drop risers are connected into a wet return or gravity drip line.

The return risers are joined into an overhead dry-return main, which is carried back to the point of discharge. The main supply riser is dripped either into the wet return or through a Webster Heavy-duty Trap or suitable size return trap into the dry-return main.

In buildings of only one story, the steam supply line is run along the ceiling to feed each radiator through a short down-feed riser which must be dripped through a return trap into a dry return. The use of the Webster Double-service Valve attached to the radiator as shown in Fig. 24-23, page 253, performs the two-fold service of supply valve for the radiator and a trap for draining the riser.

For factories, stores, loft buildings, etc., when there are a number of radiators heating one large room, Webster Modulation Valves are sometimes omitted and ordinary radiator supply valves used instead. Such systems are designated as Webster Semi-Modulation Systems to distinguish them from the usual type of modulation system.

In general, for the type of building for which the Webster Modulation System is proper, the advantage of using Webster Modulation Valves is so evident that they are considered a necessary part of the equipment.

Radiators may be exposed, concealed under window seats or behind grilles, or placed overhead to take care of skylights and unusual roof exposures, as with vacuum systems.

The radiators are drained through Webster Return Traps, into a system of return risers, and in the same manner.

Air-valves are unnecessary on the radiators, as the air is relieved through the return traps.

It should be noted, however, that as the actual difference in pressure through the supply valve and return trap of a modulation system is less than with a vacuum system, these valves and traps must not be rated as high for modulation as for vacuum system practice. It will therefore be observed, from a study of Chapter 11, that it is necessary to deduct the pressure drop for which the system is designed from the initial pressure in the heating main. With atmospheric pressure in the return piping, this difference will represent the differential pressure on which the capacity rating of the valves and traps should be based.

The products of condensation flow by gravity through the system of return risers into the basement return main, thence to a hot-well or to the receiver of a pump and receiver. If the former, a condensation pump is used to discharge the water into the boiler. In the latter case, the pump and receiver take care of the liberation of entrained air and return of condensation to the boiler.

The condensation pump, or pump and receiver, will usually be electrically driven, but if the boiler pressure is 25 or 30-lb. or above, the steam-driven type may be used.

3. Street System Carrying any Pressure: Where street steam service is maintained, the modulation system is similar in most respects to either Case 1 or 2 described above, except that no provision is made for returning the condensation to the boiler by a modulation vent trap, as in the case of a low-pressure heating boiler, or by some form of return pump where

higher pressures are carried on the boiler. The water of condensation is usually discharged to the sewer through a meter in the return line, except where a flat rate per square foot of radiation is charged in which case no meters are used.

Where exhaust steam at 1 or 2-lb. pressure is supplied by the street service, a connection is made directly from the main to the supply piping in the building. If steam at higher pressures is furnished, a pressure-reducing valve is placed between the service connections and the main heating pipe, to regulate the steam to any desired initial pressure on the system. By this means the pressure may be controlled to best suit outside temperature and weather conditions.

Webster Vacuum Systems

Webster Vacuum Systems may be sub-divided into four classes, according to the source of steam supply:

1. High-pressure or power boilers, with exhaust steam available from engines and auxiliaries.
2. Medium-pressure boilers, 15 to 50-lb. pressure.
3. Low-pressure boilers up to 15-lb. pressure.
4. Street systems.

1. Webster Vacuum System with Power Boilers: With this type of vacuum system the source of steam supply may be

(A) Exhaust steam from the engine; or

(B) Exhaust steam from engines or auxiliaries, supplemented by live steam at reduced pressure.

In the Case A when the power load exceeds the heating load, the supply of exhaust steam will be ample for the requirements of the heating system and in addition may also be used in a Webster Feed-water Heater to preheat the water supplied to the boilers. Under such conditions the heating plant is exceedingly economical since it utilizes a by-product, exhaust steam, which otherwise might be wasted.

It is under such conditions that the Webster Vacuum System is most advantageous since it ensures a rapid circulation of steam through the entire heating system with a minimum back pressure on the engine. The reduction in back pressure saves in the steam consumption of the engines.

In the Case B where the quantity of available exhaust steam is not sufficient, live steam at reduced pressure is automatically admitted into the heating main to make up the deficiency. In this design of heating plant, care should be exercised to see that all of the exhaust steam is utilized, including that from the various pumps and auxiliaries.

Fig. 15-2 illustrates a conventional layout in elevation, of a Webster Vacuum System, using both exhaust and live steam in combination with a Webster Feed-water Heater.

Referring to the illustration, the exhaust steam is made suitable for efficient heating and for subsequent use, when condensed, by passing through a Webster Oil Separator, which must be properly dripped.

It is very important that the oil separator shall be properly dripped.

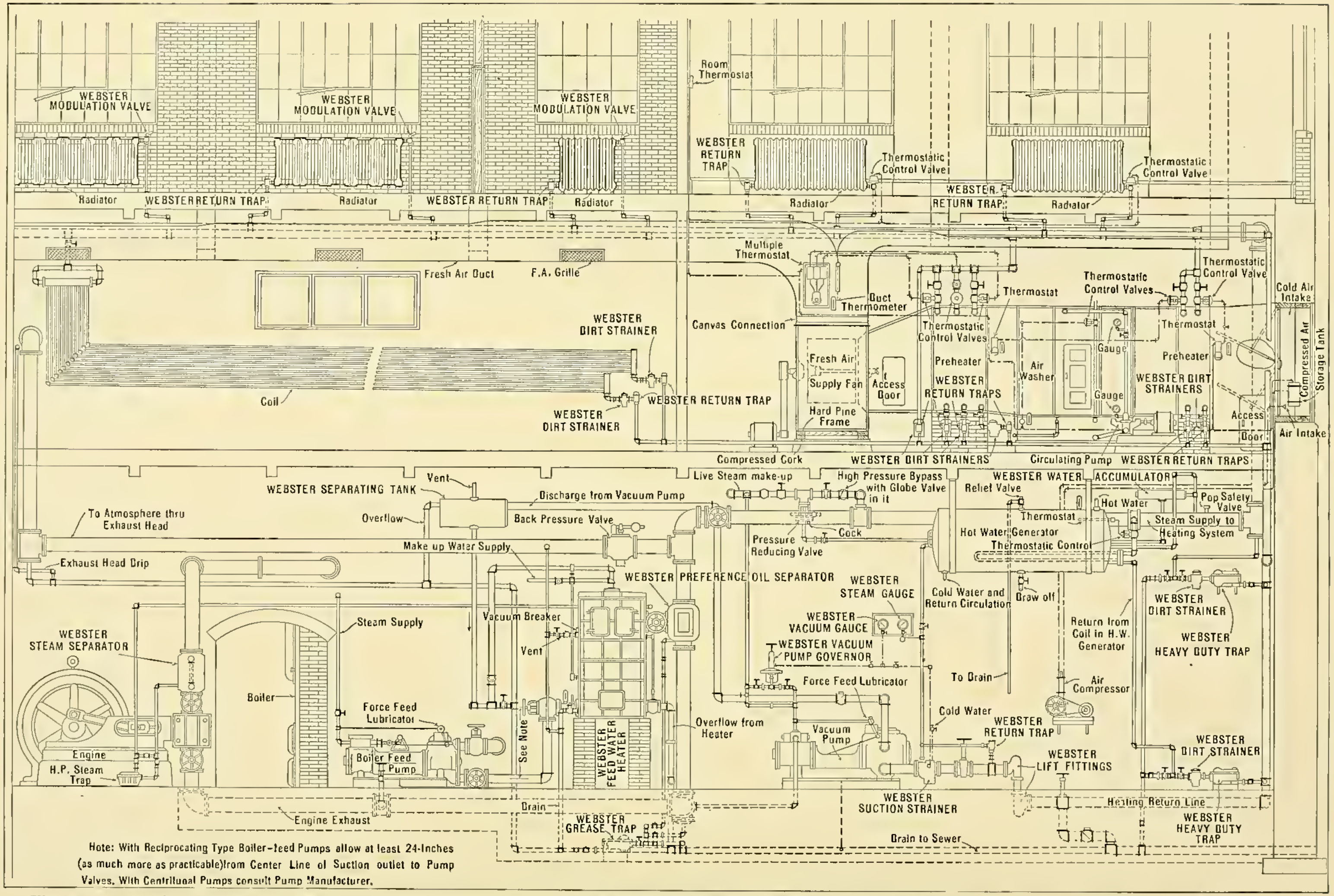


Fig. 15-2. Typical lay-out of a Webster Vacuum System for a plant generating power and using exhaust steam from engines and pumps, supplemented by live steam, as the heating medium

For ordinary cases, where the pressure in the exhaust main is maintained above that of the atmosphere, the Webster Grease Trap, also shown in the illustration, is highly efficient. A daily inspection of the grease trap should be made while the plant is in use, to be sure that it is operating properly. In systems which lie idle for a portion of the year, a careful examination should be made on starting up, to see that the grease trap and the pipe connections thereto have not become clogged on account of the solidification of the grease during the period of such idleness. Failure of the trap to function properly will cause the separated oil to be carried over into the heating system and eventually to reach the boilers, where it is very likely to produce bagging or blistering of the shell plates and tubes.

If the partial vacuum created by the vacuum pump extends into the heating mains, at times when the supply of exhaust steam is insufficient, and it is not supplemented by live steam, it will be necessary to drain the oil separator in a special manner. Figs. 24-27 and 24-28 in Chapter 24 show both methods of draining the oil separator.

The necessary live steam is admitted through a pressure-reducing valve of a suitable size and type. A Webster Water Accumulator is used, as shown in the illustration, to ensure proper functioning of the valve.. The addition of a pop safety valve in the low-pressure main, set to blow at a few pounds above the normal working pressure, will give warning of any tendency of the reducing valve to build up pressure during periods when the demand for steam is very light.

Dripping Supply Mains and Risers: Supply and return mains and risers should be sized and run as recommended for vacuum system practice in Chapter 11. The method of dripping mains and risers into the vacuum return line varies with the local conditions of each building. In the typical illustration the base or "heel" of the main supply riser is shown dripped through a Webster Heavy-duty Trap, protected from scale and sediment by a Webster Dirt Strainer.

A few general suggestions regarding the dripping of supply mains and risers will be helpful and will assist in determining which of the several methods of application will be followed.

As stated in the description of modulation systems, the overhead or down-feed system of supply piping is employed in tall buildings and in buildings of certain types where it is desirable to avoid the running of large supply mains in the basement. Steam is conveyed through a main up-feed riser to a distributing main located either on the ceiling of the top story or in the attic space above. The space should have sufficient head room to give easy access to the valves which are generally placed in the run-outs from the main to the riser, and also to permit future repairs. It is needless to say that either the attic floor should be made strong enough to carry the weight of a man or a narrow platform should be provided. Either can be made of two 2-in. thick, hard pine planks of 24-in. total width and suspended by iron hangers fastened to the roof framing and spaced at regular intervals. The platform should run parallel to pipe lines and close enough to allow a man suitable space for working.

The main riser is dripped through a Webster Heavy-duty Trap and

Webster Dirt Strainer, as shown in Figure 22-2. The drop risers are individually dripped through Webster Return Traps, with proper provision for cooling surface between the point of drainage and the trap, the surface being arranged either horizontally or vertically, as space conditions may determine. As dirt and scale are more apt to accumulate at such drip points than elsewhere in the piping system, it is essential also that the traps be protected by means of dirt pockets made up of pipe and fittings, as shown in Figs. 22-7 and 22-8, or by means of Webster Dirt Strainers, shown in Fig. 22-10. The latter are simple, self-contained fittings, easy to install, and convenient and readily accessible. The cleaning of these points where dirt accumulates is essential to the success of the heating system.

Another method of dripping the drop risers of down-feed systems, which is very satisfactory where building conditions permit its use, is to connect all of these risers into a wet-return or gravity drip line. This necessitates the running of a separate wet-return line in the basement along the floor. In such case, return traps are not needed for dripping the risers, but each riser must connect to the gravity drip line through a horizontal line in which an efficient check valve is placed. Various methods of accomplishing this are shown in Figs. 22-28, 22-29 and 22-30 in Chapter 22.

Where building conditions justify the running of a basement supply main, with a series of up-feed risers, each riser is dripped through a Webster Return Trap, protected by a dirt pocket or Webster Dirt Strainer, into the vacuum return line. The main itself is dripped at various points where it rises or where its size is reduced, so as to relieve the condensation and air which would otherwise accumulate and interfere with the proper circulation of steam. These points are also dripped through Webster Return Traps, properly protected from dirt and sediment. Provision for cooling surfaces in the pipe connection to the return trap is of prime importance with this method of dripping. (See Figs. 22-31, 22-32 and 22-33.)

Very tall buildings sometimes require a combination of the up-feed and down-feed system of supply, with a combination of the various methods of dripping.

The drip at the base of a main up-feed riser is commonly referred to as a "main riser drip" or "drip at heel of main riser." Drips at the bottom of up-feed or down-feed risers where traps are used are called "supply riser drips." Drips at various points on the basement main are called "main drips." Wet-return lines are called "gravity drips."

Supply lines to fan heater coils, hot-water generators, etc., usually require separate drips, using either Webster Heavy-duty Traps or Webster Return Traps, depending upon the volume of condensation to be handled. Where such drips are to be taken into the vacuum return line comparatively close to the vacuum pump, special provision must be made on account of the relatively high temperature of the condensation.

Supply lines to apparatus requiring steam at pressure above 15-lb., known as medium or high-pressure lines according to the pressure carried, should not be dripped directly into the vacuum return line. Special methods of taking care of such drip points must be followed. Figure 20-2, Page 203 shows one method.

Radiator Connections: Regardless of the arrangement of the supply mains and risers, and the methods of dripping them, the supply connections to the individual radiators will be similar, as shown in Figures 22-14, 22-15 and 22-19.

Horizontal connections, known as "laterals," are taken from the supply riser to the radiator. In the case of radiators with top-feed connection, a vertical supply line will be taken from the lateral to the radiator supply valve. This applies particularly to radiators of the hot-water type, in which the radiator sections are connected together at the top by means of close nipples. Sometimes steam radiators may be similarly fed, using the first section to convey the steam in a downward direction, particularly where a fractional-control or modulation valve is used with this type of radiator.

In Chapter 12 special attention is called to the necessity for proper sizing and grading of these laterals.

In Figure 15-2 the cast-iron column radiation is shown supplied through a Webster Modulation Valve, while the heating coil is supplied through an ordinary gate valve.

The advantage of the Webster Modulation Valve is that it provides a convenient, positive means of throttling the steam supply to each radiator so that the occupant of each compartment may maintain the temperature which he desires, without regard for the temperature in any other compartment. This results not only in increased comfort to the occupant, but in decrease of the amount of steam used, as the room temperature is varied by manipulation of a single valve on each radiator, and not by opening and closing windows. This latter method is the customary and inefficient way of varying room temperature where ordinary supply valves are used, owing to the inconvenience and uncertainty of such valves in throttling the supply of steam.

The Webster Modulation Valve, described and illustrated in detail in another chapter, is especially designed to give perfect modulation of room temperature with *less than a full turn of the indicator,* the position of the indicator on the dial showing the degree of opening. Further, during the period of initial warming-up of a cold room, it acts as a quick-opening valve and where the proper sizes are selected for the operating conditions, the *radiator will be heated all over in 20 minutes,* after which, if the weather conditions are such that a smaller volume of steam is required to *maintain* the room temperature, the indicator is turned back, and steam is conserved.

Radiators may be placed in exposed locations beneath windows or between columns, as shown in Figure 15-2, or may be wholly or partially concealed under window seats or behind grilles (Figs. 6-14 and 6-15); or may be located overhead as with skylight coils (Fig. 5-2).

Each of these conditions requires special arrangement of supply connections and fixtures. Some helpful suggestions to meet particular connections may be found by studying Webster Service Details in Chapter 22.

Whether to employ Webster Modulation Valves or ordinary radiator supply valves is optional with the architect or designing engineer who selects the equipment. The modulation type is recommended wherever efficiency and economy of operation are desired, as the additional first cost of installation is very little, and repairs and upkeep are negligible.

They are especially to be recommended in hotels, apartment houses and other buildings with transient occupants who have no incentive to economize in the use of steam where ordinary valves are used. Also greater economy may thus be secured in dormitories, schools, institutions, etc., where the manipulation of the radiator valves is under control of a regular attendant rather than the occupant of the room. For such cases, a lock-shield type of Webster Modulation Valve with key is frequently used.

The Webster Vacuum System is admirably adapted for use where special systems of automatic temperature control are used, as in large office buildings, hotels, etc., to control individual room temperatures.

Disposal of the Products of Condensation: The air, gases and water comprising the products of condensation of steam within the radiators, are drained from each radiator by a Webster Return Trap connected at the return end. Lateral "run-outs" conduct this condensation to a series of return risers which convey it to a system of basement return mains, in which a partial degree of vacuum is maintained by a steam or electrically driven vacuum pump, according to conditions.

The Webster Return Trap serves the triple function of relieving the air and gases as well as the water of condensation and also preventing the escape or loss of steam into the return line.

Air valves are unnecessary. Their annoyances and discomforts are entirely eliminated.

The several types of Webster Return Traps and the various methods of application for different conditions are explained in other chapters.

As with laterals from supply risers, return run-outs to risers must be properly sized and graded. This is a detail which often requires personal inspection during the progress of the installation, particularly where the laterals and run-outs are run in pipe or sheet-metal sleeves which in turn are embedded in concrete or other solid floors.

The Vacuum Pump: The vacuum pump and its auxiliary equipment may be referred to as the heart and lungs of a vacuum system. It is all-important that they be properly selected and sized, and that the function of all parts of this equipment be thoroughly understood so that the piping connections will be properly made. (See Chapter 13.)

Various types and arrangements of equipment are necessary to meet different conditions.

In the type of vacuum system which is now being described, the vacuum pump will usually be of the steam-driven reciprocating type, steam being furnished directly from boilers at relatively high pressure.

The supply of steam to the pump is automatically controlled by a Webster Vacuum-pump Governor actuated by the degree of vacuum existing in the vacuum return line and adjusted to stop or slow down the operation of the pump as the vacuum approaches the point for which the governor is set, and starting or speeding up the pump as the vacuum drops below this point.

The pump, where of the reciprocating type, is lubricated by the admission of cylinder oil into the steam supply line through a sight-feed lubricator. or if preferred, through a mechanical force-feed oiler, the latter being attached

to the pump preferably before shipment and actuated by the operation of the pump itself.

The suction valves of the pump are protected from dirt and foreign material by a Webster Suction Strainer.

The products of condensation will be conveyed by gravity through the system of return risers and main vacuum-return line to a point either above or below the suction inlet of the pump, depending upon building conditions.

If this point is below, the vacuum pump will raise the condensation with its entrained air. The arrangement of "lifts" depends upon the vertical distance and degree of vacuum created and maintained by the pump.

Webster Lift Fittings used in pairs will materially assist the vacuum pump where lifts are necessary.

Various methods of applying vacuum-governors, lubricators, suction strainers and lift fittings in connection with vacuum pumps are shown in the Webster Service Details in Chapter 13 in which the practical problems of installation are worked out.

Final Disposal of the Condensation: The vacuum pump discharges the products of condensation to a point of disposal, where the entrained air is liberated and the condensation returned to the boiler as feedwater.

In Figure 15-2 the pump discharges into a Webster Receiving Tank which is vented to the atmosphere. The condensation flows by gravity from the tank to the Webster Feed-water Heater against the working pressure carried.

In the typical case, the receiving and air-separating tank is of the water-control type, and the Webster Feed-water Heater also has an automatically controlled valve in its water-supply line.

As the water level in the Feed-water Heater lowers, the automatic valve opens, and the condensation flows from the tank to the heater through the sealed connection. This arrangement of tank and heater may be used only where the tank can be located at sufficient height above the heater so that the static head will overcome the working pressure within the heater.

Additional fresh water required to make up any losses that occur is admitted automatically into the tank by the lowering of the water level, which in turn actuates the automatic water-regulating valve.

Surplus condensation overflows from the tank to the sewer or drain. The waste of condensation at higher temperature from the overflow of the feed-water heater is thus eliminated.

An alternate arrangement which is often desirable is the use of a Webster Receiving Tank of the plain type with a Webster Feed-water Heater of the Steam-control Type, as is shown in Fig. 27-7, Page 304.

In this case the condensation flows continuously from the tank to the heater. As the water level in the heater rises, the automatic valve, placed in the steam line to the boiler-feed pump and actuated by the water level in the heater, causes the pump to withdraw the water from the heater.

Another arrangement is the use of a Webster Tank of the plain type discharging into a special return inlet on the heater, fresh water as needed being automatically admitted into the heater. (See Fig. 27-6, Page 303.)

Still another arrangement which is necessary where the tank cannot be

located at sufficient height above the heater to overcome the pressure therein, is the use of a Webster Hydro-pneumatic Tank, described in Chapter 13.

Where an open feed-water heater is not used, the tank discharges to the boiler-feed pump, either the water-control or steam-control type of pump being used, according to conditions.

The specific functions of each of these types of Webster Receiving Tanks are more particularly described in Chapter 24.

Ventilation Problems: In Figure 15-2 a typical installation of a motor-driven ventilating fan, with its re-heater and tempering coils, is also shown.

The fan heater supply line is dripped through a Webster Return Trap and Dirt Strainer, and the individual heater sections through Webster Return Traps.

The method of dripping fan heater sections will vary with the size, arrangement and number of sections. Special study should be made of the various Webster Service Details shown in Chapter 22.

It is exceedingly important not only to choose the right type of trap for use with indirect radiators but also to have the pipe connections properly made. The trap must be of the highest efficiency, with sufficient capacity to pass rapidly the maximum quantities of water and air which are present when first warming up, and afterwards open for the condensate and entrained air but absolutely prevent the escape of steam. This must be done even where core sand and greases are present and settle in the valve bodies. Where groups of radiators are made up of large numbers of sections nippled together, there is a likelihood of air-binding sometimes extending over considerable areas. This trouble can be avoided if the traps and piping are right. Webster Return Traps and Webster Heavy-duty Traps meet every condition if installed in accordance with proper Service Details.

Further reference should also be made to other chapters for description and method of application of various types of Webster Feed-water Heaters where power boilers are used for generating steam for prime movers; Webster Steam Separators placed in the high-pressure steam lines to provide dry steam for engines; and Webster Expansion Joints, of both the single and double-slip pattern, for low and high-pressure steam lines, to take care of the expansion and contraction which occur in such lines.

2. Webster Vacuum System With Medium-Pressure Boilers, 15 to 50-lb.: The foregoing description will serve as a general description of this type of vacuum system, except that the feed-water heater will not be used, the exhaust steam will be limited to that from pumps and auxiliaries, if steam-driven, and the vacuum pump will be either of the low-pressure steam-driven type or electrically driven.

Under some conditions, particularly for pressures up to 20-lb., electrically driven pumps may be more suitable, and in these cases the lubricator and vacuum-pump governor will not be used.

For boiler pressures up to 15-lb., either electrically operated reciprocating vacuum pumps or steam-driven pumps can often be used in conjunction with Webster Hydro-pneumatic Tanks to return the water to the boiler without the use of a separate boiler-feed pump. Webster Service Details in Chapter 13 show the proper arrangement for such cases.

3. Webster Vacuum System With Low-Pressure Boilers, up to 15-lb.: The description of this type of vacuum system is the same as that immediately preceding, except that the rotary type of electrically driven vacuum pump, handling air and water separately, is particularly suitable. These pumps also act as boiler-feed pumps if the conditions of the plant are within the range of the discharge head or pressure at which the manufacturers guarantee these pumps to operate.

4. Webster Vacuum System. Steam Furnished from Street System: As steam at a pressure suitable for operating a steam-driven vacuum pump is usually not available, this type of vacuum system will require either rotary or reciprocating electrically driven vacuum pump.

The condensation in such cases is discharged to the sewer or point of disposal through a condensation meter of a type for vacuum service.

Webster Vacuum Systems, Special Modifications: There are two special types of modifications of Webster Vacuum Systems which will require special description: The Webster Conserving System and the Webster Hylo Vacuum System.

Webster Conserving System: This is a special modification of the Webster Vacuum System which meets two general conditions:

1. Where necessary to operate steam-driven pumps from low-pressure boilers at very low pressure—from 5 to 20-lb.

2. Where necessary to provide steam for some special service *continuously* at a pressure higher than that needed for heating.

Referring to Figure 15-3, this system is in general respects similar to

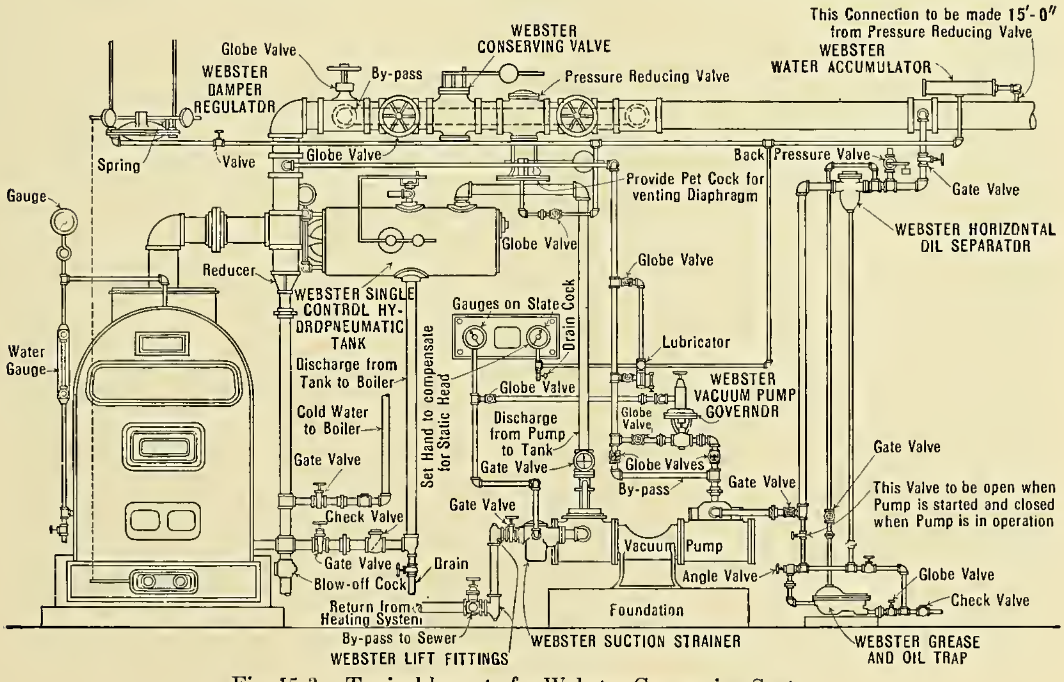


Fig. 15-3. Typical layout of a Webster Conserving System

the vacuum system described for working pressures from 15 to 50-lb. pressure.

A low-pressure steam-driven vacuum pump is used, discharging to a Webster Hydro-pneumatic Tank, and thence to the boiler against pressure.

The distinguishing feature of this special system is the Webster Con-

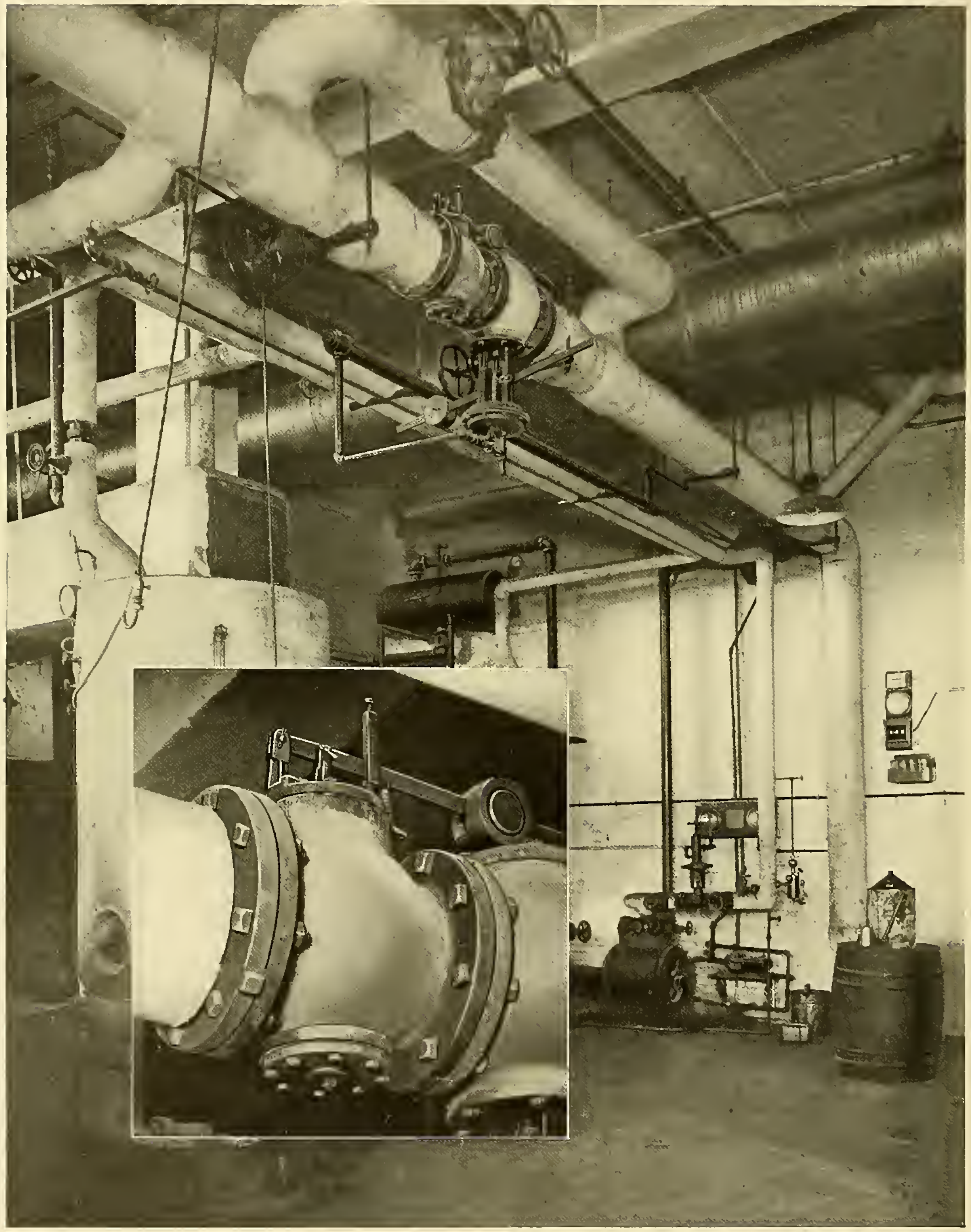


Fig. 15-4. Typical installation and close-up of the Webster Conserving Valve

serving Valve, which is placed in the supply main near the boiler, and conserves or retains the steam on the inlet side of the valve until sufficient pressure has been built up to (1) operate the pump, or (2) meet the pressure requirements of the special service.

Connections to the vacuum pump or for the special service are taken from the high-pressure side of the conserving valve. When the predetermined pressure has been built up, the excess pressure is released into the heating main by means of the conserving valve.

In consequence, the vacuum pump begins to function before the steam enters the heating main and continues to operate even when the pressure drops on the high-pressure side to such point that the conserving valve closes against further admission of steam into the heating main. The heating system is therefore kept continuously drained of water at all times, insuring return of condensation to the boiler and preventing accidents or damage which would occur from lowering the boiler water level to a dangerous point.

One other special feature of this system is the use of a Webster Damper Regulator to control the boiler pressure, operating from the low-pressure side of the conserving valve. The damper regulator must be connected in the special manner recommended.

In a similar manner to the above, any special apparatus like kitchen equipment requiring steam continuously at higher pressure is always assured of constant supply regardless of operation of the heating system.

Another adaptation of the Webster Conserving System is in large plants in which the engines are run condensing.

A study of steam engine performance, where the engine exhausts into the atmosphere or into the heating system aga nst a back pressure slightly above that of the atmosphere, shows that engines working under such conditions actually convert only 5 to 10 per cent of the heat supplied to them into mechanical energy. The remaining 90 per cent of the heat originally supplied to the steam entering the engine is retained in the exhaust.

In some plants, power and heating loads are nicely ba anced so that all the exhaust steam available from power units can be utilized for process work or heating purposes, in which event the 90 per cent of heat energy remaining in the exhaust steam is put to useful work. In such cases the engine may be considered as a pressure-reducing valve which reduces the pressure from that carried on the boilers to that required for heating and process purposes.

There are numerous industrial plants where the power load is greatly in excess of the heating load, so that the quantity of exhaust steam available is greatly in excess of that actually required. The surplus exhaust steam with its heat units must then be wasted.

Where these conditions exist, the engines are often operated condensing instead of non-condensing, so that exhaust steam from the auxiliary machinery only is available. In most instances the quantity is not sufficient to supply the heating load, and the deficiency is made up by live steam supplied from the boiler through a pressure-reducing valve.

The work done by the pressure-reducing valve in reducing the steam from boiler pressure to that required in the heating system is converted into superheat on the low-pressure side of the valve. This work represents

about 10 per cent of the total heat energy supplied to the steam. If this 10 per cent of heat energy can be utilized by conversion into mechanical energy, nearly ideal conditions will be approached.

Various attempts have been made in the past to improve the economy of power and heating plants by endeavoring to utilize the exhaust steam from the receivers of compound engines. This exhaust is bled into the heating system and the deficiency made up by admitting live steam into the receiver through a pressure-reducing valve. In determining the advisability of this form of application, the effect of the relations between heating and power load and the relative proportion of the cylinders so vitally affects the economy that in each instance special consideration has to be given to all elements entering.

The Webster Conserving System can be applied to this problem. In the same manner that the conserving valve is applied to conserve the pressure on the boiler by preventing the escape of its steam until a certain predetermined pressure is obtained, it can be applied to the receiver of a compound engine, opening and admitting steam at receiver pressure into the heating system, when the pressure on the receiver exceeds that which is necessary for the proper operation of the low-pressure cylinder, and closing when the receiver pressure drops below the point for which the conserving valve is set.

The quantity of steam taken from the receiver is made up by changing the cut-off on the high-pressure cylinder so that the high-pressure side acts as a pressure-reducing valve for the steam required for heating purposes. In expanding from boiler pressure to the receiver pressure, the heat energy given up in the expansion is converted into useful mechanical energy.

By means of the Webster Conserving System many existing power and heating plants may be brought to efficiency where they are otherwise wasteful of steam.

Webster Hylo Vacuum System: Where a number of buildings must be heated from a detached central plant, or where a building covers considerable ground, the source of steam supply and of vacuum cannot always be located to make a well-balanced system.

The largest building in the group may, for various reasons, be farthest from the source of supply, and may also be the lowest point in the system of return piping, thus making it doubly difficult to secure perfect heating and easy return of condensation. Nearby points may be favored with unnecessary pressure difference.

Attempts have been made to solve this problem by running the supply and return mains in reverse direction, so that the point of highest pressure is the point of lowest vacuum and inversely, thus maintaining, in some degree, the same differential between supply and return pressures.

Where the largest building is at a low point away from the source of supply, it is obviously impracticable to solve the problem in this way. Furthermore, such a plan does not allow for extensions to or expansion of the plant, unless the new buildings can be located to suit the piping scheme, irrespective of the manufacturing need.

This problem has been solved with unqualified success by Webster

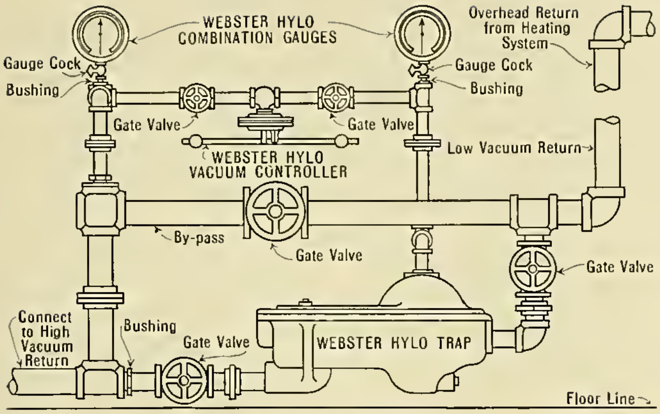
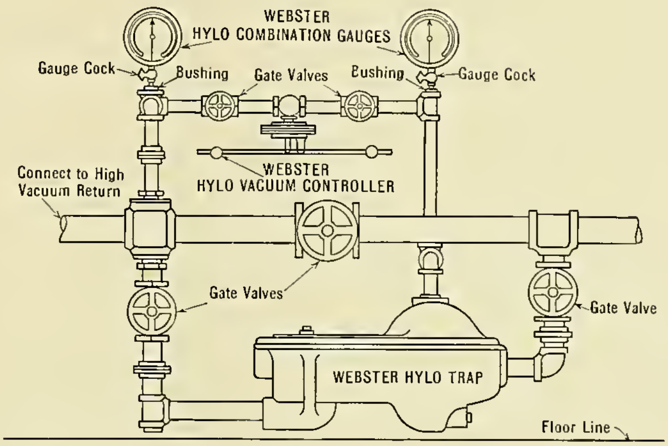


Fig. 15-5. Connections around Webster Hylo System equipment where the low-vacuum return main drops from overhead and discharges through a Webster Hylo Trap to the high-vacuum return main

Fig. 15-6. Typical installation of Webster Hylo Trap, Controller and Gauges where high and low-vacuum returns are on the same level



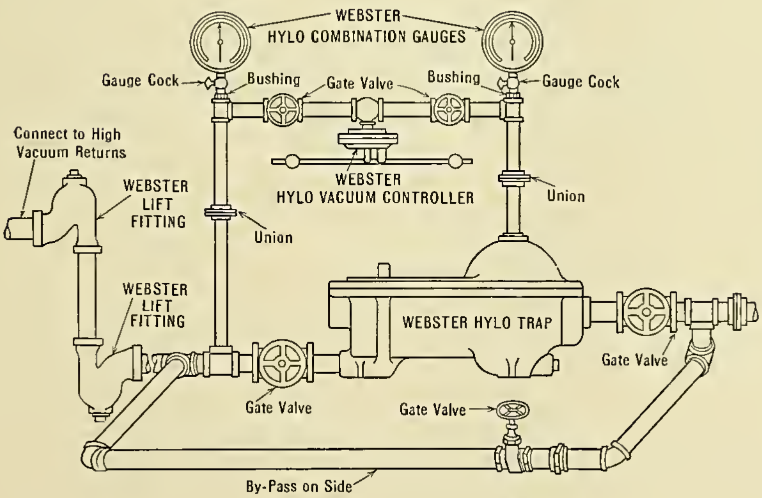


Fig. 15-7. Arrangement of the Webster Hylo Controller, Trap and Gauges where the low-vacuum return is lifted to the high-vacuum return

Hylo Vacuum Controlling Sets, which are installed at certain points in the return line to restrict the vacuum to just the amount necessary for proper circulation and drainage at nearby points where high vacuum is not needed. The high vacuum is carried to extreme or low points where high vacuum is required. The result is a well-balanced system with perfect circulation in all parts.

The operation of the vacuum pump is also improved to a marked extent as the degree of initial vacuum is reduced, making it unnecessary to use or waste cold water to condense the vapors arising from the hot water returned under high vacuum. Sometimes smaller pumps may be used, or the pumps may be operated at slower speed with less wear and tear.

The Webster Hylo Sets consist of a Webster Hylo Trap, a Webster Hylo Vacuum Controller, Webster Hylo Vacuum Gauges, and when needed, Webster Lift Fittings.

Figures 15-5, 15-6 and 15-7 show various methods of connecting Webster Hylo Sets to meet different building conditions.

CHAPTER XVI

Application of the Webster System to Lumber and Other Kiln Drying Problems

PROPER seasoning and drying of raw lumber is a first essential to well-finished products in any wood-working industry.

This basic condition makes the dry kiln or room a most important feature, for as proved by experience in many instances, lumber that was found defective when worked would have been satisfactory if proper methods had been applied for drying. Very careful attention should therefore be given to the design of the drying room, the character of apparatus used and the heating medium employed.

The method to be employed in drying will depend entirely upon the condition of the product when put in the kiln. Green lumber, or lumber having a high percentage of moisture, will require a different method of procedure, and a longer time to dry than lumber which has been air dried. Hard woods such as oak or hard maple usually require a longer time than soft woods.

Saw mills should determine the percentage of free moisture by test and so mark each pile of lumber when first piled in the yard. Later, when it is sold, the lumber should be tested again and the two records given to the factory or other purchaser.

Factories should test and mark the lumber when first received, and if it is piled in the yard to be kiln dried later, it should be tested before going to the kiln and again before removal, these records being placed on file.

The process required for the drying of lumber in kilns is properly divided into four parts, as follows:

First: The primary treatment, during which all dampers are closed, 100 per cent humidity is maintained and the stock is warmed through without drying.

Second: The initial drying period, during which the conditions of temperature and humidity within the kiln are advanced sufficiently to reduce the moisture content to 25 per cent.

Third: The intermediate drying period, during which drying conditions are still more advanced to reduce the moisture content to 10 per cent.

Fourth: A final drying period, during which extreme conditions are used to further reduce the moisture content to the percentage desired.

Improper drying methods will usually result in one or more of the following conditions:

(1) Percentage of moisture not correct for working, (2) case hardening, (3) hollow-horning or honey-combing, (4) molding.

The operator should make careful test readings to determine the moisture content both before and during the drying of the lumber.

Records from such tests will give data on which to base his treatment of the stock. Tests should be made at stated intervals of 48 to 72 hours

during the drying period. For this purpose test boards from which samples may be taken should be inserted in the kiln. A good solid heavy piece as a sample, or better still, two or more sections out of as many different boards taken out of the pile one-third the distance from the bottom, will yield an average or representative test for moisture content. With two or more tests for moisture showing varying results, it is safer to use readings showing the highest moisture content rather than the average of the pieces.

At the same time, tests should be made for case hardening. If the lumber becomes case hardened, it practically stops the drying process, or at least slows it to a great extent. Frequently this results in hollow-horning, cupping, internal strains and many other evils which affect the stock throughout the manufacturing process.

Almost all "working" which occurs in furniture, or other wood articles, is due to stresses which developed in the wood during the seasoning period. These stresses may be determined by two simple tests and eliminated before the stock leaves the kiln.

The manufacturers of the different makes of dry kilns furnish detailed instructions for the various tests on which the successful operation of their kilns depend.

The final condition of the lumber required in different factories varies with the purpose for which the lumber is used. For instance, in wagon work, many manufacturers do not use lumber containing less than 10 to 12 per cent of moisture; in auto body work, for open bodies, 6 to 8 per cent is considered proper; for closed bodies, 5 to 6 per cent. Furniture manufacturers generally dry down to 4 to 6 per cent, while wheel manufacturers dry the spokes as nearly bone dry as possible, but do not dry the felloes below 8 per cent, the theory being that when the wheel is made the spokes may absorb moisture and make a snug fit.

A modern kiln is usually constructed with brick side walls and a roof of tile or cement covered with roofing felt, tar and gravel. The doors are of special design to allow for easy loading and unloading, and to prevent, as much as possible, air leakage and loss of heat. Ventilating flues are provided in the side walls for supplying air and removing same as desired.

The heating medium usually employed is steam at varying pressures, depending upon the kiln temperature desired. The temperature within the kiln is controlled by means of a thermostat operating a valve in the pipe supplying steam to the coils.

A system of steam spray pipes is provided under the material to be dried for increasing the humidity as desired and to assist in warming the stock. The percentage of humidity in the kiln may be automatically controlled by means of a humidistat operating a valve controlling the supply of steam to the spray pipes.

Where steam, whether exhaust from engines and auxiliaries, or taken direct from the boilers, is used as a heating medium, the success of the drying equipment depends upon the manner of carrying this steam to the heating units, the proper drainage of the supply mains, the circulation of the steam through the heating units and the removal of air and water of condensation.

All manufacturers of drying equipment utilizing steam as a heating medium recognize the importance of these features. One of the largest manufacturers of drying equipment in the United States says in its book of instructions:

"Where troubles have been experienced, investigations have shown that they are generally due to one or more of the following conditions:

"Poor steam service.

"Pressure not constant.

"Wet steam due to improper condensation drainage.

"Insufficient steam pressure.

"Poor drainage from traps.

"Improper design of supply and drainage piping.

"Traps allowing steam to blow through into the main drainage line, holding back kiln drainage.

"Traps on heating units not functioning properly.

"Traps stopped with scale or dirt.

"Trouble is often caused by faulty design in making steam connections to kilns.

"All steam lines must pitch in the direction of steam flow. Automatic drain traps must be provided at all low points on these lines in order that there may be absolutely no condensation lying in the lines at these places, and that steam may enter the kiln dry and at a high temperature. Failure to provide proper methods of drainage will result in reduced volume and temperature of steam and correspondingly low temperatures and poor service in dry kilns."

The important features in connection with the steam supply and drainage system can be enumerated as follows.

(1) Adequate and continuous supply of steam. Pressure of steam constant and sufficient to produce the required temperature within the kilns.

(2) Manner of conveying steam to coils.

(3) Method of draining main steam supply.

(4) Character of design of heating units.

(5) Method of complete and rapid air removal from heating units and from entire return system.

(6) Method of removal of condensation from heating units.

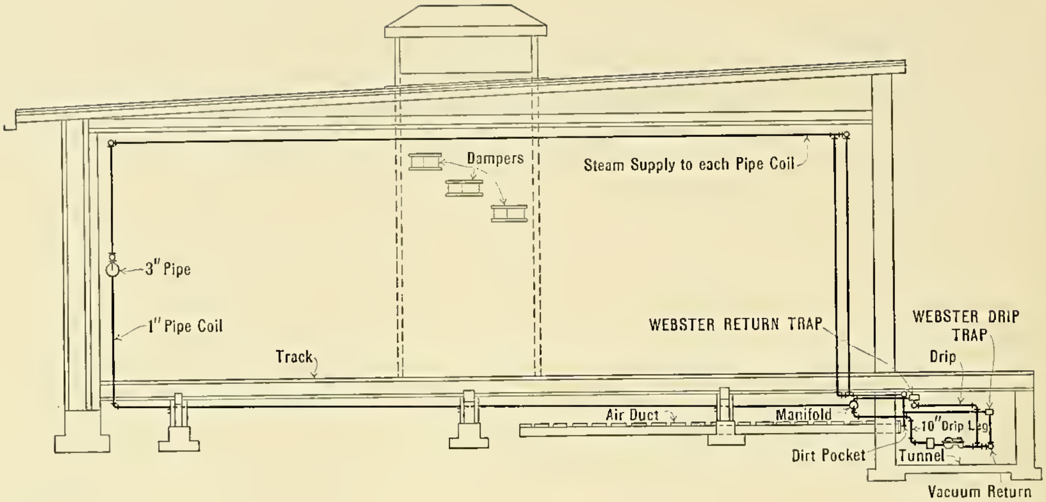
(7) System of drainage piping.

(8) Ultimate disposal of water of condensation and of air.

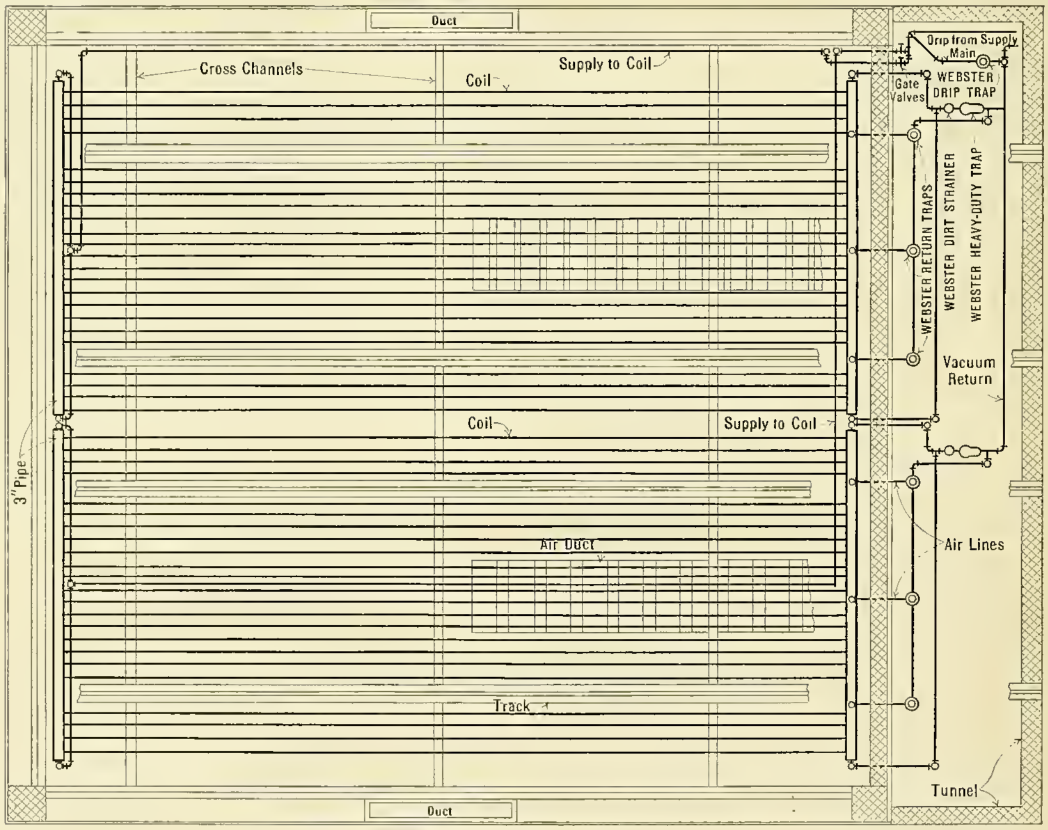
(9) Adequate and continuous pitch of pipes throughout the entire length of the coil.

Items one, seven and eight will be governed materially by the conditions existing at the plant where kilns are to be used, and as these conditions vary with the character of the plant, this discussion will be limited to the requirements of the kiln only.

The pressure of steam supply, so far as the operation of the kiln is concerned, will depend upon the temperature required within the kiln. If a maximum kiln temperature of not more than 150 deg. fahr. is required, satisfactory results can be obtained by the use of exhaust steam from engines and auxiliaries at a pressure not to exceed 1½-lb. gauge. The same results



A Typical Elevation



A Typical Plan

Fig. 16-1. Sections through a typical dry kiln with coils of the continuous-header type using Webster Heavy-duty Traps for drainage and Webster Return Traps for removal of air from return headers

will be obtained, of course, by using steam direct from the boiler, reduced to a corresponding pressure by means of reducing valves. It is very important to place a relief valve on the low pressure side of the reducing valve to prevent rise of steam pressure to a point where there is a liability of injuring the thermostatic return traps. The details are shown in Fig. 22-3, Page 216.

Where temperatures greater than 160 deg. fahr. are required it will be necessary to increase the pressure of the steam accordingly. In good practice the temperature of the steam must not be less than 60 degrees higher than the temperature desired in the kiln.

The size of the steam supply mains will depend upon the volume of steam to be delivered, and the drop in pressure allowable. This may be determined with the help of the tables in Chapter 11 in this book after a decision has been reached as to the total heat requirements of the kiln and the distance of the kiln from the source of steam supply. The same principles apply for the installation of steam mains to the kilns as would apply for the installation of steam mains for any other purpose.

Extreme care should be given to the drainage of the steam main at the point of entrance to the kiln. It is advisable that water of condensation from the main shall be relieved from the bottom into the return and that steam for kilns shall be taken from the top of the main rather than to allow the condensation to drain through the coils. The supply main may enter the kiln from a point above the coils used for heating, or from below them.

Manufacturers of drying equipment have devised numerous types of heating units but practically all have standardized on those constructed of pipe. The coils are placed either vertically along the side walls of kiln, or horizontally in a space provided underneath the material to be dried. In the latter instance they are usually installed in a horizontal position, although some manufacturers prefer coils placed vertically. The advantage of more equal heat distribution is claimed for the large unit laid horizontally, but this is not fully realized unless the removal of air and condensation is complete.

With coils having short vertical headers, say 10 pipes high, it is very important to secure an equal distribution of steam to all of the pipes. The internal diameter of the supply header should be ample; 2½-in. is none too great. It is very important not to locate the inlet in such a position that steam will enter those pipes directly in front of it and passing through to the return header, tend to pocket the air in the other pipes. The removal of air will be very sluggish and meanwhile the efficiency of the whole coil will be low. A deflector placed within the header in front of the inlet will improve the steam distribution. A much better method is to have more than one inlet. These additional supply connections will also reduce materially the velocity of the entering steam.

Figs. 22-21 and 22-22, on Page 220, show methods of splitting up the return header into two parts, for coils of more than 10 pipes, when there is a liability of air binding.

With horizontal headers, particularly where of some length, the internal diameter should be large and the number and location not only of supply openings but also of return and air vent outlets should be selected

with great care, so as to ensure uniform distribution of steam and complete removal of air and water.

Practical experience has demonstrated that incomplete removal of air and condensation has caused unequal heat distribution throughout the kiln as well as a drop in temperature of from 20 to 50 per cent. The air must not only be removed from the coils but also must be rapidly and completely eliminated from the return system and discharged outboard.

The selection of the proper type of trap to be used in any given case depends upon the steam pressure which it is necessary to carry on the coils to secure the requisite heating effect, the quantity of water which the trap must handle, the temperature of the room in which the trap is installed, the pressure in the discharge line and the disposition to be made of the products of condensation.

A continuous and uniform steam pressure of not over 3 to 5 lb., a moderate and uniform quantity of condensation to be handled, and a temperature of not over 80 deg. in the space where the traps are located, are the most favorable conditions for the successful operation of low pressure thermostatic traps. *They should not be employed where the temperature requirements of the kiln necessitate carrying a continuous steam pressure which approaches closely the allowable maximum pressure of the trap. Traps on high*

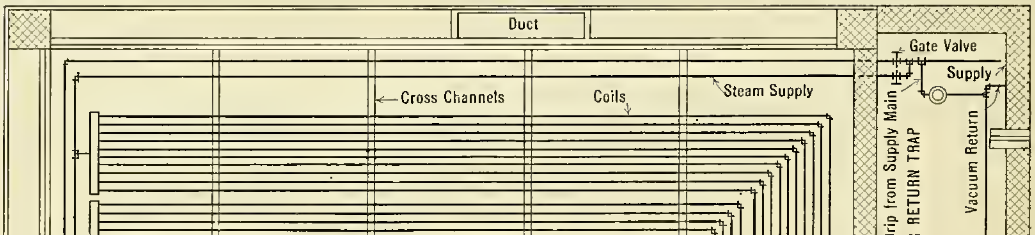
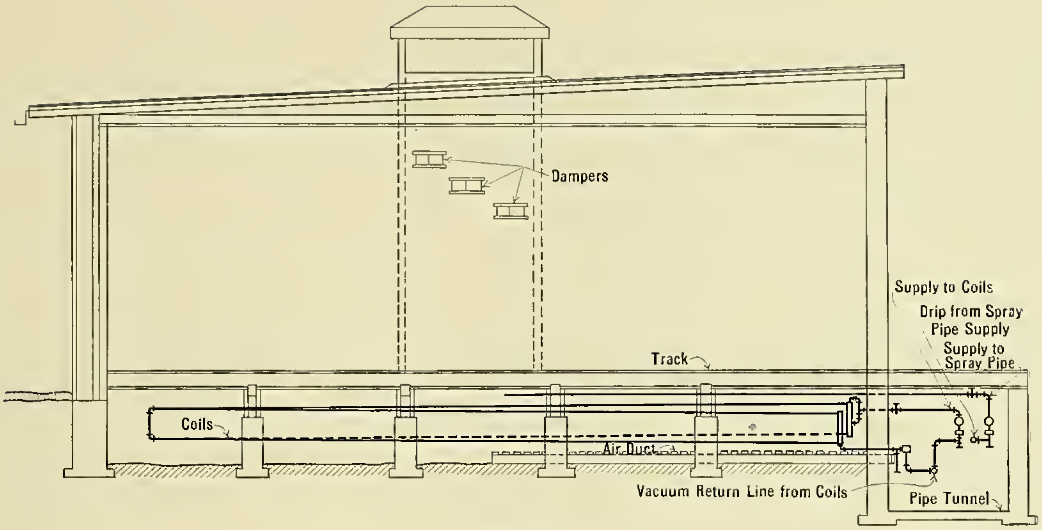


Fig. 16-2. Typical section through a dry kiln using coils of the sectional-header type



Elevation

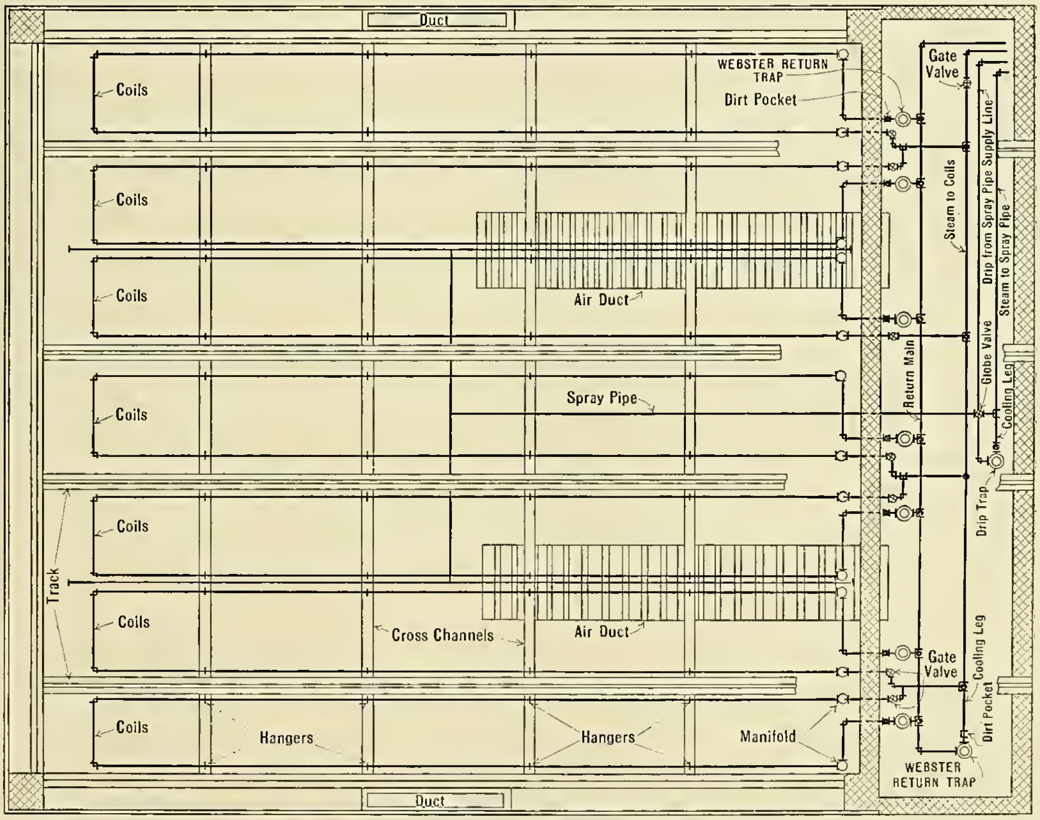


Fig. 16-3. Sectional drawings of a typical small dry kiln using individual return traps for drainage of coils

pressure steam drips must never be permitted to discharge directly into the return pipe near the thermostatic traps on account of the liability of back pressure or water hammer. The connection should be made at a point beyond the traps, placing a check valve in the thermostatic trap return to prevent back pressure therein and in addition means should be employed for disposing of the high temperature vapor as shown in Fig. 20-2, Page 203.

The types of heating units which are universally used and the manner of applying the Webster specialties for proper air removal and drainage of condensation are shown in Figs. 16-1 to 16-4 inclusive. Attention is called to the importance of providing a dirt strainer for the drain connection to each trap. Both the traps and strainers should be readily accessible. *Where thermostatic traps are used they should be located where they will not be affected by the high temperatures of the kiln.* This is usually accomplished by extending drain connections to the extreme front or rear of the kiln and placing the traps near the floor.

On small units as shown in Figure 16-3, where thermostatic traps are used, additional provision for the removal of air is unnecessary, but where a large volume of condensation accumulates, additional provision for air removal is essential and heavy-duty traps should be used. Where the heating unit is of the continuous header type as shown in Figure 16-1 the air removal can be accomplished by the use of heavy-duty traps equipped with a thermostatically actuated air bypass within the trap and by means of additional thermostatically actuated air traps connected into the top of the main return header, as shown in Figures 16-1, 16-2 and 16-4. The number and location of these air traps is governed by the length and design of the main return header. The outlets of these air return traps should be connected into the main vacuum return line beyond the discharge connection of the heavy-duty trap.

Where heavy-duty traps are used there should be a drop leg of from 8 to 10 inches between the outlet on the return header and the trap inlet.

Where it is desired to drain the condensation from two or more coils to one heavy-duty trap, or where the return header of the coils is of special construction divided into two or more sections and the condensation from all sections is drained by one trap, it is essential for the proper removal of air to equip each return header, or each section of the return header, with a thermostatically actuated return trap. The outlets of these traps should be connected into the main vacuum return line in the same manner as described above.

Pipe coils and return pipe connections to traps must have a sharp downward pitch their entire length in the direction of the flow of condensation. The coil supports must be of a permanent character and so arranged that any subsequent settlement of the kiln structure will not affect the pitch of the pipes.

The discharge from all heavy-duty traps and thermostatically actuated return traps used in connection with kilns may be connected into a common return line, but it is preferable that this return line from kilns shall be extended independently from the kilns to the vacuum pump, rather than to connect it into returns from the heating system of the manufacturing

plant or other equipment. The condensation rate from the kilns will fluctuate, depending upon the temperature within the kiln, the nature and condition of the product being dried and the outside temperature. Consequently, at times when the air removal and condensation rate from the kilns is high, trouble may be experienced with the operation of other equipment if connected to the same return line. Also, if the same efficient equipment is not used in connection with the heating system or other equipment, as is used in connection with the kilns, the poor operation of the heating system or other equipment will naturally reflect in unsatisfactory operation of the kilns.

The amount and location of radiation installed within the kiln will depend upon the location of the kiln, the temperature desired within the kiln, the steam pressure, and nature of product to be dried. This constitutes a special branch of engineering and engineers thoroughly familiar with this class of work should be consulted.

The method for figuring the total radiation required by a given dry kiln will not vary from the descriptions given in detail in Chapter 5, except that during the warming-up period an additional heat factor is required to care for the moisture content of the lumber or other material being dried.

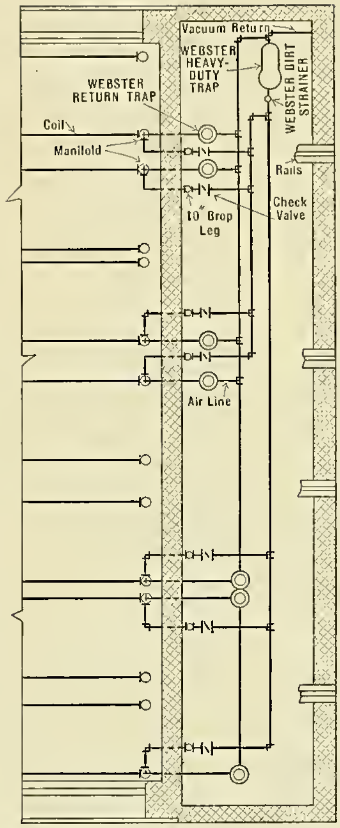


Fig. 16-4. Showing the connections where two or more coils are drained through one Webster Heavy-duty Trap

Much of the general information on lumber drying was furnished for this Chapter by the National Dry Kiln Co., of Indianapolis, Ind.

CHAPTER XVII

Application of the Webster System to Slashers and to Cloth and Paper-drying Apparatus

SLASHERS are used in the textile industry for sizing and drying warps or yarns before they are placed in looms to be woven into cloth. In these machines, steam is supplied usually to two cylinders, of 5 and 7 ft. diameter, over which the yarn passes to be dried after sizing.

Ordinarily the steam supply and the drainage connections are on opposite heads of the cylinders, the connections passing through the cored shafts upon which the cylinders revolve. Steam is carried through the mains to the slasher at about 80-lb. pressure and before it enters the cylinders is reduced to between 5 and 12-lb. per sq. in. by a pressure-reducing valve. The steam pressure in the cylinders of course always must be above that of the atmosphere as the rapid drying of the materials demands that the surface temperature of the cylinders shall be above the atmospheric boiling point.

Owing to the light weight of the metal used in the construction of slashers, vacuum breakers, usually three in number, are provided in the head of the discharge side of each cylinder. These open when a partial vacuum occurs in the cylinder and prevent collapse of same.

The condensation is raised to its point of removal from the slasher by means of troughs or buckets, usually three in number, attached to the inside cylindrical surface. A pipe attached to each bucket carries the condensation to the hollow cylinder shaft and thence through the bearing to the outside. From there the condensation goes through the Webster Traps, etc., to the point of disposal.

The Webster System for draining slashers provides the most efficient drying effect with least attention to the drainage equipment. It has succeeded in overcoming entirely the frequent delays and slowing down of the manufacturing processes previously experienced with other devices.

As will be seen in Figure 17-1, each cylinder is equipped with a Webster Return Trap, a Webster Dirt Strainer and a bull's-eye sight glass.

The Webster Return Trap permits the free passage of air and water and closes against the discharge of steam. The Webster Dirt Strainer protects the trap from dirt and the sight glass enables the operator to see whether or not the drainage system is functioning.

A bypass is provided around the drainage apparatus. When starting up, the bypass may be opened for a few minutes to permit the quick discharge of air. After starting, the slasher is drained automatically through the Webster equipment.

A pressure sufficiently above that of the atmosphere must be carried in the cylinder to dry the goods and this is sufficient to discharge the condensation and air through the Webster Trap, if free vent to atmosphere is maintained. There is no advantage in connecting the discharge of the traps

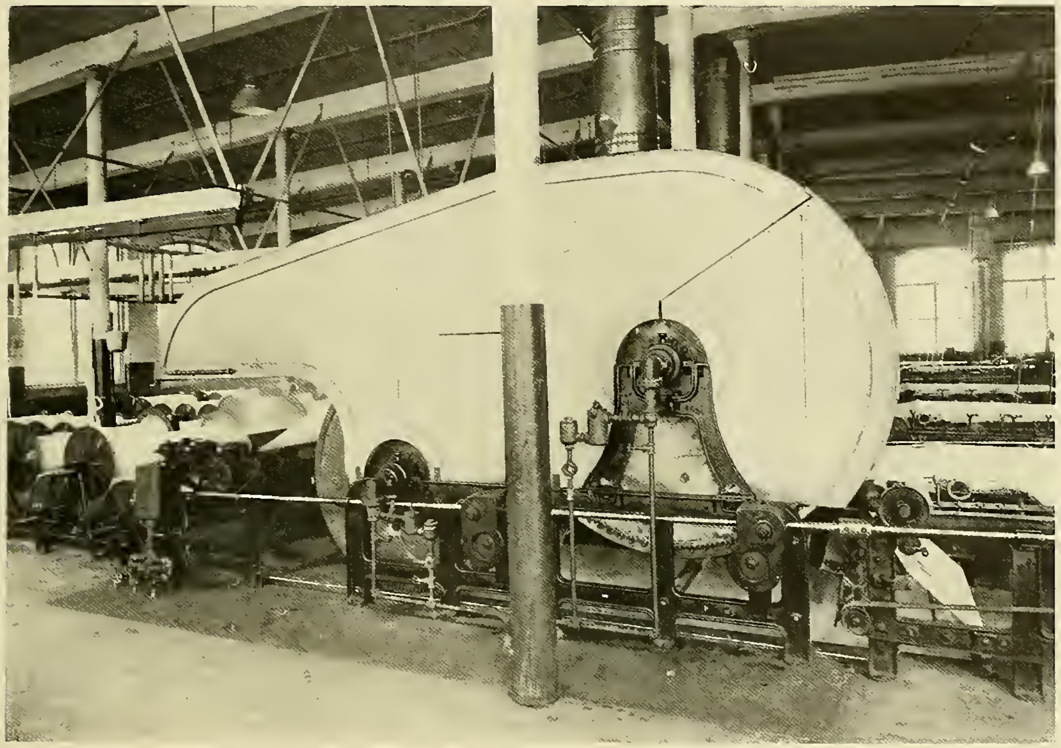
to a vacuum pump if sufficient vertical distance is available to allow a proper fall for the condensate to flow by gravity to an open receptacle.

The condensation rate with this type of slasher will vary from 400 to 600 lb. per hr.

One of the best-known American manufacturers of slashers states in his catalog:

"We strongly recommend the use of Warren Webster & Co.'s apparatus for slasher drainage.

"Steam traps can be furnished if desired but we recommend the use of the Webster System in preference, as higher economy will certainly maintain



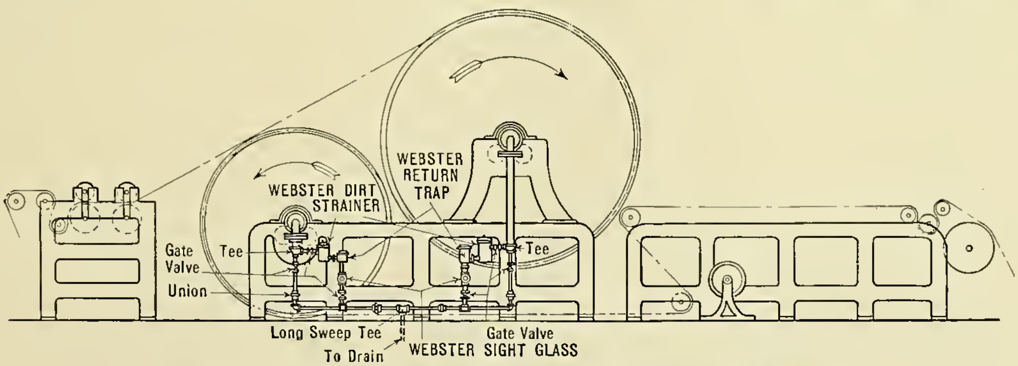


Fig. 17-1. Typical application of Webster Apparatus to a slasher

a higher rate of production and its simplicity lessens the liability of stoppage to which a system of steam traps is apt to be subject after a few years of use.

" The Webster System as compared with a steam-trap system insures steady, instead of intermittent, drainage and practically an entire absence of condensation in the cylinder with all consequent advantages."

Cloth and Warp Drying Machines: Except in details, the process of draining drying machines of both vertical and horizontal types is the same as for slashers.

Each cylinder is provided with troughs or buckets which, as the cylinder revolves, empty through a pipe to a hollow shaft and through the journal to the return duct.

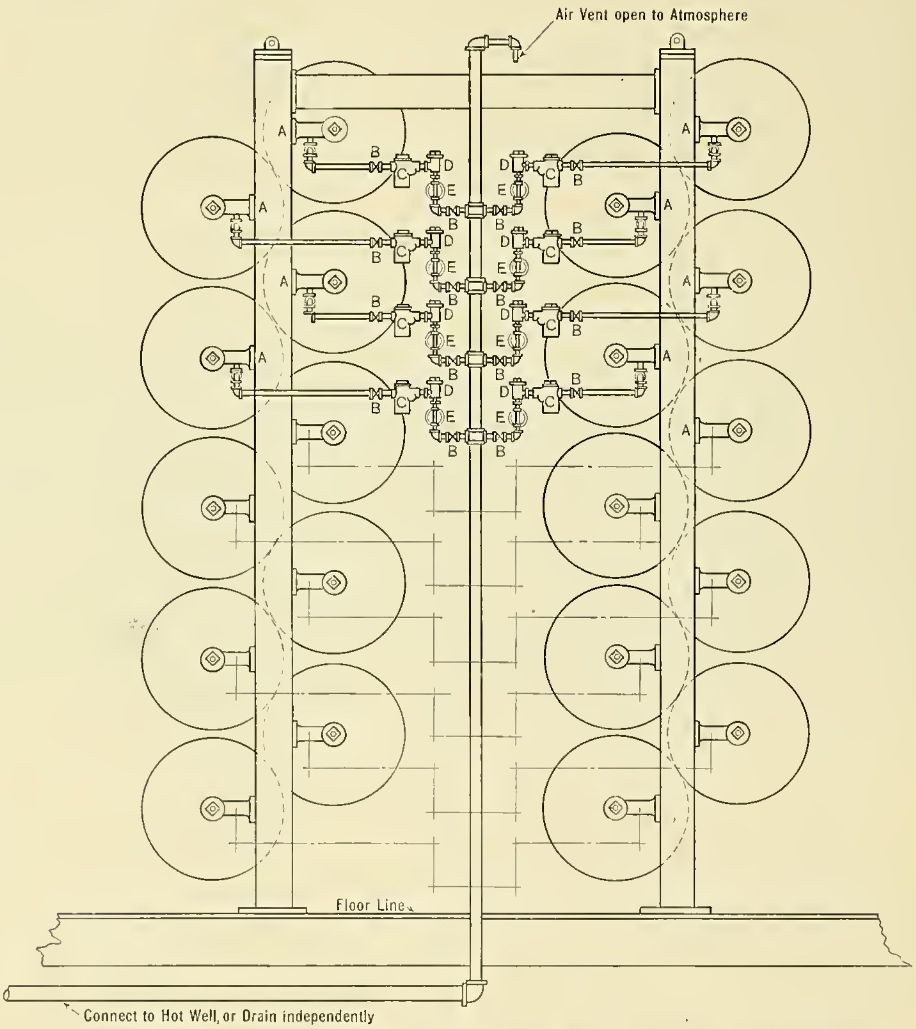


Fig. 17-2. Application of the Webster Apparatus to a vertical drying machine

A. Solid copper gasket inserted between bracket and housing. A copper gasket having hole equal in area to that in the bracket must also be placed between the bracket and housing on the inlet side to keep cylinder alignment true. B. Gate valve. C. Webster Dirt Strainer. D. Webster Return Trap. E. Webster Bull's-eye Sight Glass

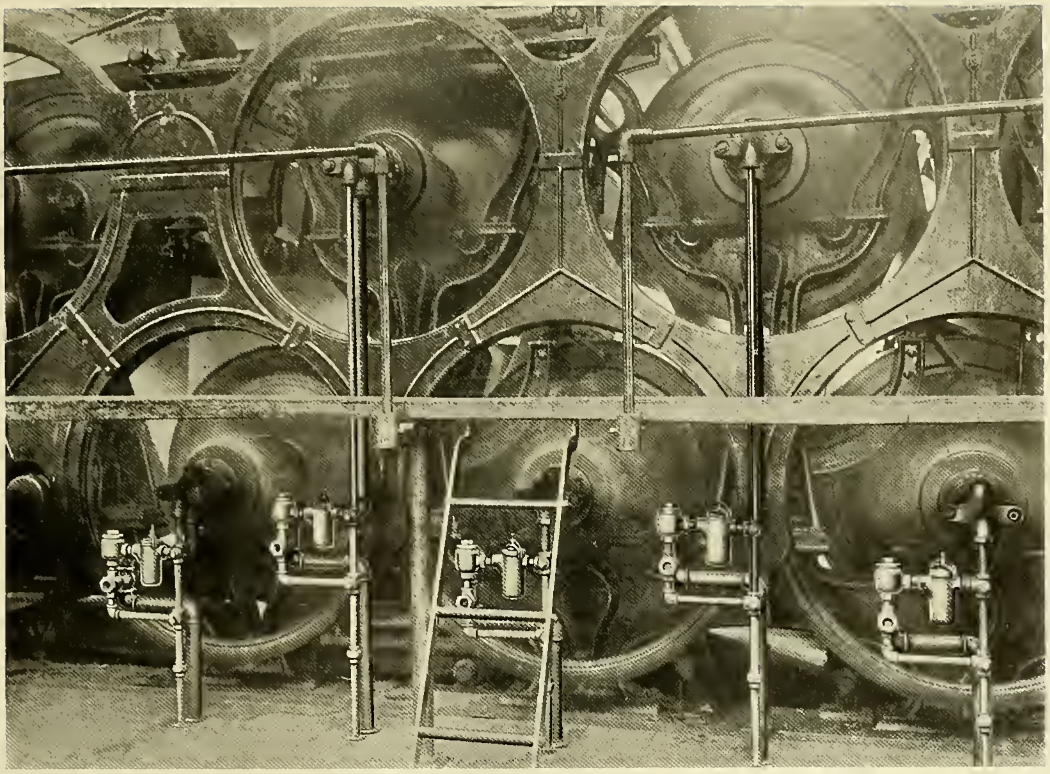


Fig. 17–3. Application of Webster Apparatus to paper machines where there are separate drips for each cylinder

The housings of the machine and the brackets supporting the cylinders are cored to provide ducts for conveying steam to the cylinders and condensation away from them. The frame on one side acts as a supply pipe while that on the other side acts as a return. Steam at a pressure of 15 lb. per sq. in. or less is admitted to the housing and passes through the

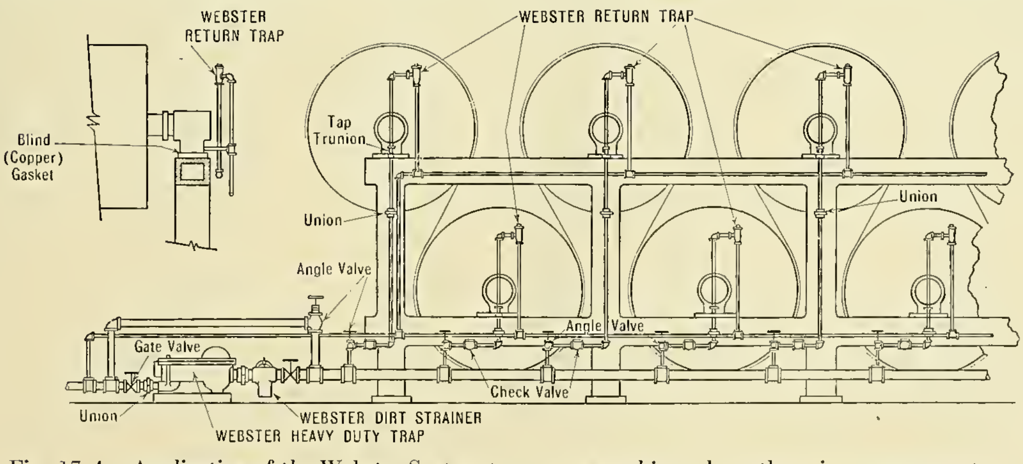


Fig. 17–4. Application of the Webster System to a paper machine where there is a common return line for all cylinders with air removed separately from each cylinder

brackets and the journals to the cylinders. To prevent collapse, vacuum breakers are installed in the cylinder heads, usually on the discharge end.

Frequently it is advisable to make two or three separate steam supply connections to each housing, as the area of the cored opening in housing is too small to convey the required amount of steam without too great a pressure drop.

The duct in the housing through which the products of condensation pass can best be drained by the use of one or more Webster Heavy-duty Traps provided with thermostatically controlled air by-pass.

PAPER MACHINES: Two types of machines of particular interest are used in the manufacture of paper, cylinder machines and Fourdrinier machines. Both require the evaporation of large quantities of water from the paper after the pulp has been pressed and the web has formed.

After passing through the presses the paper usually contains about 45 per cent of water. This moisture is reduced to about 5 per cent, depending upon the thickness of sheet and the finish desired, by passing the paper over a series of drying cylinders, the inside surfaces of which are heated by either exhaust or live steam at low pressure or a combination of the two.

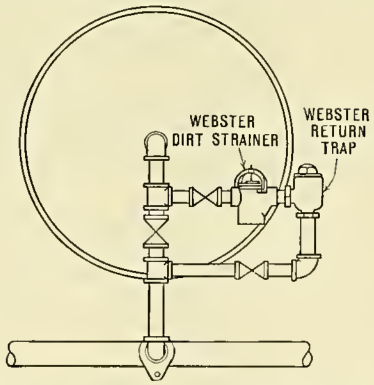


Fig. 17-5. Method of draining cylinder of a paper machine using Webster Return Trap and Webster Dirt Strainer. These connections are suitable for operation with either vacuum or gravity discharge

Usually the steam-supply header runs parallel with the machine, close to the floor, a hole being bored in the header and connected by a pipe to the cored journal on the cylinder.

The return header runs either above or below the steam header and has the same kind of connections as the supply.

The drying cylinders vary in size and length. For the purpose of removing the water, one type of cylinder is equipped with buckets and another with what is termed a siphon pipe. Cylinders equipped with buckets discharge the condensation only when in motion, while those equipped with siphon pipes discharge whenever water accumulates, provided there is sufficient pressure in the cylinder or vacuum in the return line to give the necessary differential.

The condensation per square foot of exposed drying surface of the cylinders depends upon the speed of operation and the thickness and width of the paper on the cylinders. The stock from which the paper is made, together with the amount of water extracted by the press rolls, also has a direct bearing upon steam consumption. The condensation will average about $1\frac{1}{2}$ lb. per sq. ft. of total roll surface and naturally is greatest at the wet end of the machine.

The drainage from the cylinders may be removed either by gravity or by means of a vacuum pump, whichever is desirable.

Usually with the Webster System of drainage, a Webster Return Trap

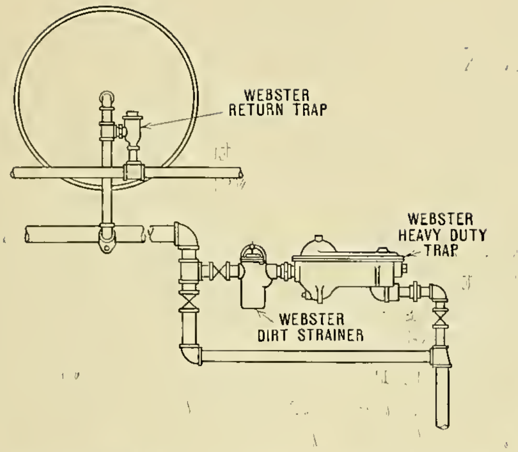


Fig. 17-6. Method of draining cylinder of a paper machine using Webster Heavy-duty Trap and Webster Dirt Strainer

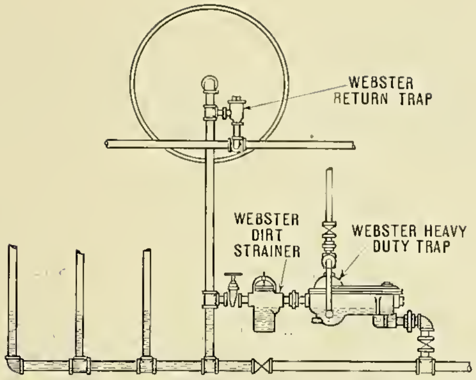


Fig. 17-7. Method of draining cylinder of paper machine for gravity discharge where a water line is to be maintained, using Webster Heavy-duty Trap with balanced steam connection, Webster Dirt Strainer and a Webster Return Trap for vent discharging into dry returns

with its Webster Dirt Strainer and bypass is provided for each cylinder as shown in Figures 17-3 and 17-5. All traps discharge into a main return which leads to the point of disposal, which is a feed-water heater or hotwell, open to the atmosphere for the removal of air.

Webster Heavy-duty Traps are sometimes used instead of Webster Return Traps (Figure 17-6) especially where the presence of a water line is desirable in the return (See Figure 17-7).

The reader is referred to Page 184 for a complete discussion of the selection of the proper type of trap and the precaution which should be observed where thermostatic traps are used.

CHAPTER XVIII

Application of the Webster System to Railroad Terminals and Steamship Piers

THERE are many uses for thermostatically actuated return traps where the pressures carried are greater than in heating-system work. Instances involving operation under steam gauge pressures of from 15 to 100 lb. are described in this and following chapters.

The requirement, in all cases, is that the return trap shall discharge the water and air of condensation without waste of steam and that the fixture being heated shall be maintained at maximum efficiency.

In these special installations, certain fundamentals must be observed to secure successful operation. *The first requires that the thermostatically actuated traps must discharge directly to the atmosphere or to a return line in which atmospheric pressure is maintained.*

This latter condition may be obtained by venting the return line free to the atmosphere. In some cases the same result is secured by discharging the returns into a flash tank, the vent of which is connected to the low-pressure heating main, while the condensation is cared for through the usual type of return traps to the vacuum return.

Railroad Terminals—One of the greatest causes of delay in the daily movement of hundreds of trains into and out of terminals where there is freezing weather is the difficulty in keeping switches clear of snow and ice.

Many terminals have therefore adopted the method (Figure 18-1) of placing steam-heating coils between the ties, under the switches. Due to the unusual exposure, these coils and their supply lines are operated under 60 to 80-lb. gauge pressure in order to prevent freezing. The dripping of

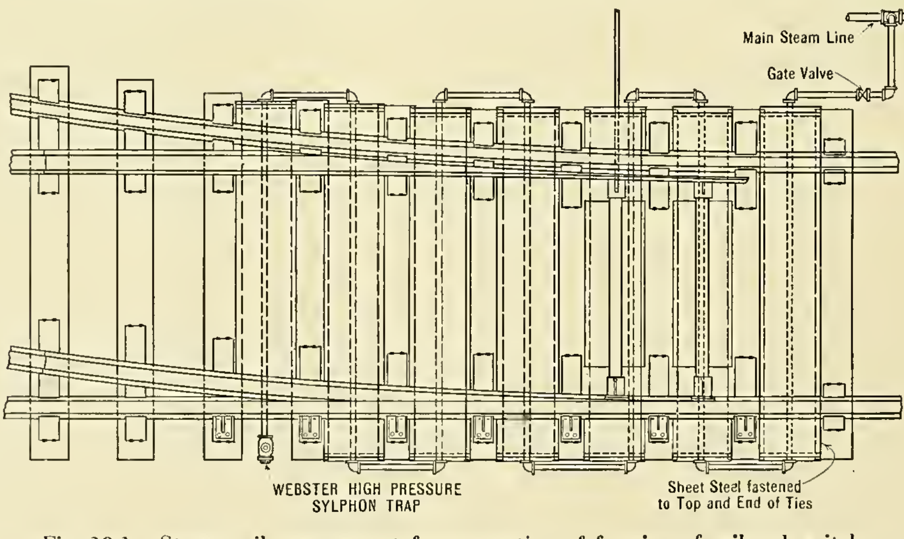


Fig. 18-1. Steam coil arrangement for prevention of freezing of railroad switches

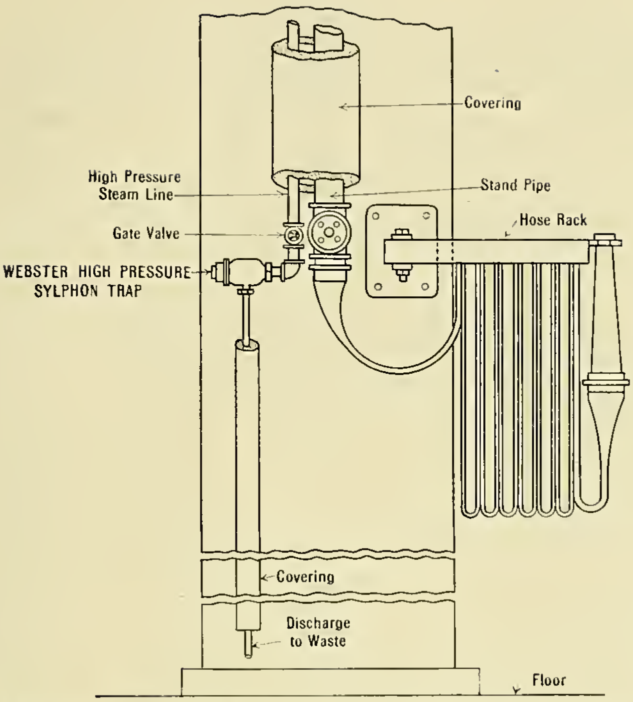


Fig. 18-2. Method for prevention of freezing of fire protection lines. The water and steam pipes are encased in the same insulation and the steam pipe is drained by a thermostatic return trap

these lines and coils presents a double problem: *First*, water and air of condensation must be freely discharged onto the road-bed, and *Second*, condensation must not form steam clouds that might obscure nearby switch signals.

A type of thermostatically actuated return trap which answers these requirements has been developed by Warren Webster & Company after many tests and experiments. This return trap is fitted with Monel-metal seats and valve pieces to withstand the wire-drawing effects of steam at high pressure differential. The thermostatic member is placed on the outboard or atmospheric side of the trap, and as the trap is generally placed in the rock ballast of the road bed, its exterior is usually given a special finish to give it protection against the elements. (See Page 275.)

Railroad terminals are also equipped with extensive systems of water lines for fire protection purposes and these lines, too, must be kept from freezing. The method of prevention (Figure 18-2) found most satisfactory is to run a steam line, carrying from 60 to 80-lb. gauge pressure, parallel with and close enough to each water line that both steam and water lines can be encased in the same insulating covering. Where the water lines terminate, as at hydrant and hose gate outlets, the same dripping of the steam lines and the same thorough removal of condensation with absence of steam cloud are required as with the yard switches.

The same type of return trap is used in both cases.

Steamship Piers: Steamship piers in cold climates are somewhat similar to railroad terminals in that the fire lines must be protected. In addition, heat is required for a large number of small enclosures scattered throughout for housing the pier clerks.

Piers are so built that water of condensation from coils heating water lines and clerk houses cannot be easily returned. The practice is to discharge the condensation overboard through the pier deck. The return traps must, therefore, keep the lines clear of condensation to avoid possibility of freezing and at the same time avoid waste of uncondensed steam.

Webster Return Traps of similar construction to those previously described for railroad terminals are successfully used for this work.

CHAPTER XIX

Applications of the Webster System to Vacuum Pans and Similar Apparatus

IN processes of manufacture where boiling of the product at a low temperature is desirable, a special application of the Webster System has been devised for removing air and water of condensation.

One of the important uses for vacuum pans is in the milk-condensing industry and in the following pages this particular application of the Webster System is discussed. However, the principles and the Webster apparatus are equally applicable to other processes such, for instance, as the manufacture of sugar, salt, candy or tartaric acid.

The development and growth of the milk industry has reached a point in the last few years where it is now necessary, due to keen competition, to use not only the most modern and efficient machinery in the process of milk treatment, but to install modern power equipment and a perfect system of steam circulation in order to insure the commercial efficiency of the plant.

It is essential that each pound of steam (live or exhaust) shall do the maximum of useful work and that all water of condensation shall be returned to the boiler.

There are numerous uses for exhaust steam in the modern condensory, such as heating of boiler feed water, heating of water for general use and in the heating system of the building, but as a rule these require only a small portion of the amount of steam available from the exhausts of the engine, compressors, pumps, etc.

In a condensory of say 100,000-lb. capacity of milk daily, there will be available at least 200 hp. of exhaust steam, not over 20 per cent of which is required for any of the above uses. The remaining 160 hp. of exhaust steam is available for use in the vacuum pans.

The usual practice in the past has been to use live steam in the heating coils of the vacuum pan at a pressure of about 15 to 20-lb. gauge, reducing to this pressure from the high-pressure mains. Very often excess exhaust steam from the engines has been wasted to the atmosphere, being considered a by-product of the engine room with little value excepting for its uses in the boiler room. Exhaust steam at 5-lb. gauge pressure contains about 88 per cent of the heat content of the live steam used to develop power and is as effective in the vacuum pan coils as live steam reduced to the same pressure.

To make use of exhaust steam at 5-lb. gauge pressure where live steam was used in the vacuum-pan coils, only slight changes are necessary. Occasionally the sizes of coil connections must be increased to the size of the coils themselves and where the steam pressure is decreased, a slight additional amount of heating surface in the coils will be required on account of the lower temperature of the steam at this pressure. In some plants where exhaust steam has been substituted for live steam without changes in the

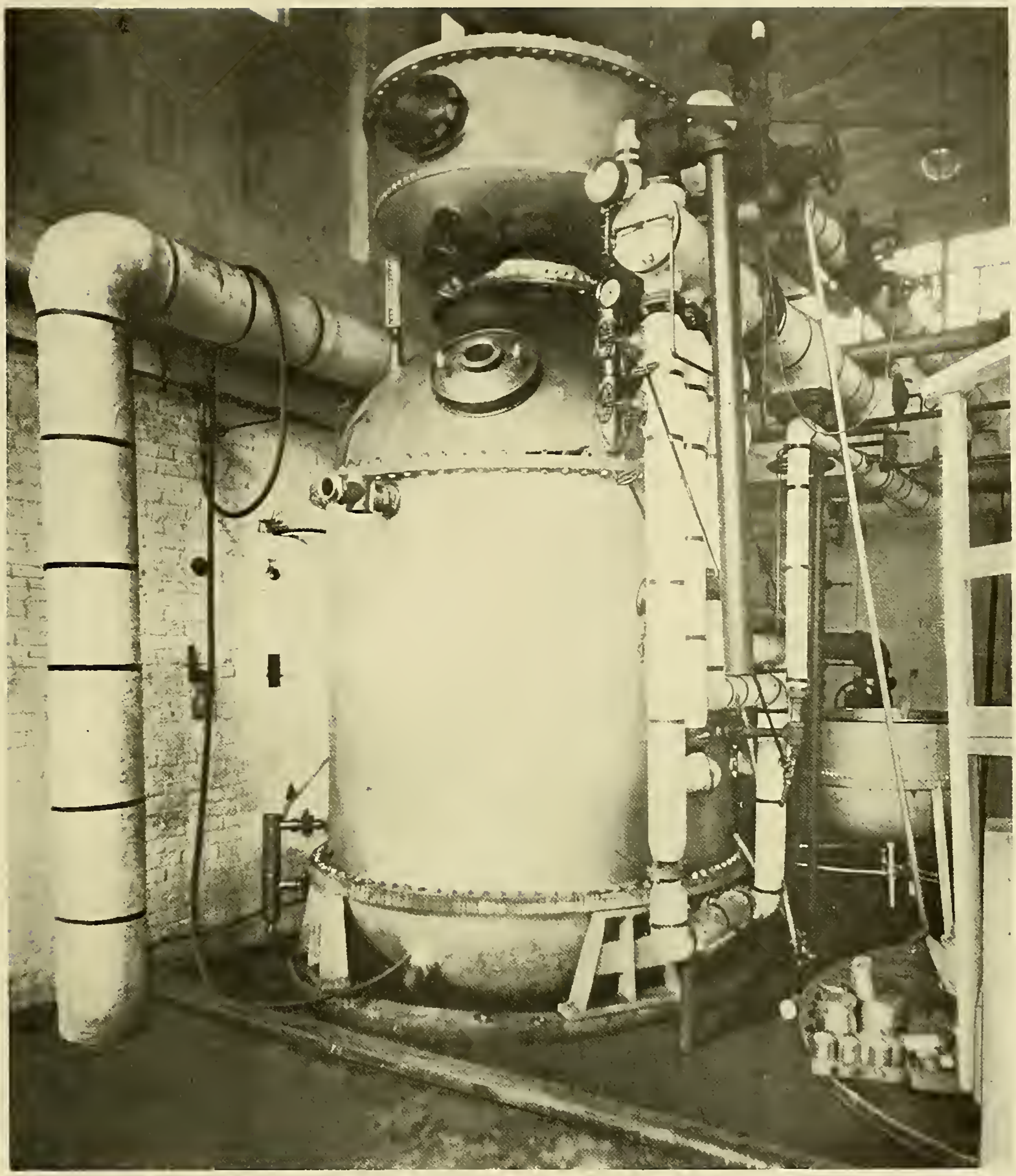


Fig. 19-1. Milk condenser

heating surfaces, a slight additional time was required to condense the batch of milk. In most cases this increase was not more than ten minutes.

The usual control valve connections, that is, the double globe valve and a gauge attached to each coil connection, will be the same for use with the exhaust steam as with the live steam.

The return connections for use with the exhaust steam are very simple. A single Webster High-differential Heavy-duty Trap (see page 249, Chapter

24), with a bypass, is connected to each coil outlet. These traps discharge to the return main leading to a vacuum pump in the boiler room. It is essential that each coil shall be drained separately into the vacuum return main in order that the pan operator may have absolute control of the steam pressure in each individual coil.

It is necessary when condensing milk to vary the pressure in these coils at will. In some instances the pressure in certain coils must be reduced to atmosphere, while the pressure in other coils is increased to as much as 5 lb. per square inch in order to cause a positive circulation of milk within the pan. Without this positive control of circulation it is impossible for the pan operator to properly manipulate the process.

It is also imperative that the water and air of condensation shall be removed immediately from the coils of the vacuum pan and that this shall be accomplished independently of any conditions which may affect the operation of the general exhaust steam system in the plant.

It is advisable to use an independent pump and return line for the vacuum pans and not to depend upon other similar equipment which may be used for heating the building. The return line should have a gradual gravity pitch to the vacuum pump and should be so arranged with by-passes and valves that in case the vacuum pump should become inoperative for any reason the return condensation may be discharged by gravity. There must necessarily be no pockets of any nature in this return line.

A maintained vacuum of 6 to 8 in. at the outlet of the trap is usually sufficient to insure at all times a positive circulation of steam and the instant removal of all water and air of condensation.

Not only are much better results obtained by the certainty of this circulation, but in many cases where exhaust steam has been substituted for live steam in the milk-condensing process, a marked improvement in flavor of the product has been noted.

The great saving in steam consumption in a condensory when equipped with the Webster System will usually pay for the entire installation within a few months. However, a careful analysis must be made of the existing conditions of an old plant or the requirements of a new condensory before any exact arrangement can be determined. There is no other single improvement to a condensory that will approach the saving obtainable through the economical use of exhaust steam.

Figure 19-2 shows an older type of connection for vacuum pans, in which high-pressure steam only is used. The pressure is reduced from 125-lb. per sq. in. boiler pressure to 15 or 20-lb. per sq. in. for use in the pan.

The outlet connections are pipes without valves or checks, leading to a header which is piped to a tank located beneath the pan. The tank is a receptacle for water and air of condensation. The air is vented through the small vent valve while the water is drained to a high-pressure positive return trap which discharges the water to an open hotwell or to a feedwater heater.

The difficulties encountered in this construction will be short-circuiting of the steam from one return to the other and the impossibility of maintaining independent or separate pressure control on each coil in the pan.

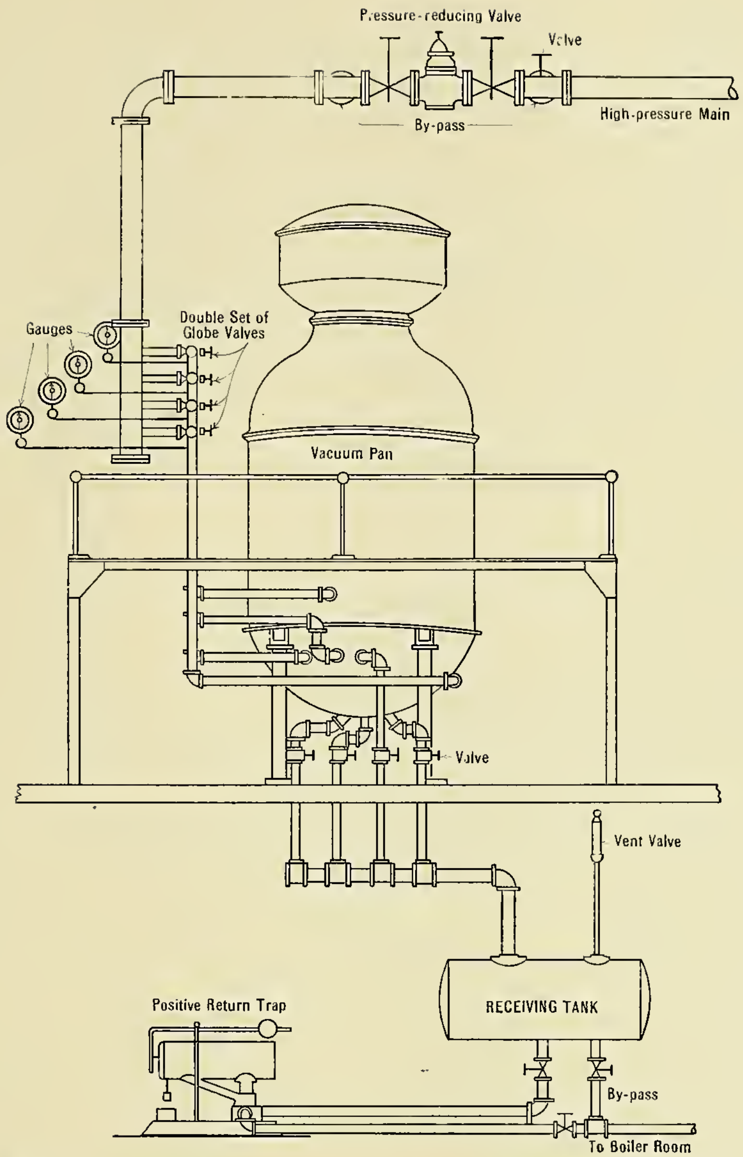


Fig. 19-2. Drainage system for a vacuum pan using a positive return trap and receiving tank

The system of piping, however, is in common use in most of the smaller condensories at the present time.

Figure 19-3 shows another construction where the inlet connections are similar to those in Figure 19-2, but where the outlet connections are controlled by means of gate valves and check valves which discharge into a common return line. This return line is run direct to a pump and receiver which discharges the water back to the boiler. A great many installations are somewhat similar to this and it is evident that there is a great deal of

waste of steam due to the inability of the operator to properly throttle the controlling valves on the outlet connections.

Figure 19-4 shows the approved application of the Webster System.

The exhaust-steam piping includes a Webster Steam and Oil Separator and an auxiliary connection from the high-pressure main with pressure-reducing valve. It is essential that the pressure-reducing valve shall be of such construction that it will maintain constantly the pressure which is desired when it is necessary to use live steam for condensing. The back-pressure valve must be of such construction that it is impossible at any time to exceed 10 lb. per sq. in. pressure on the low-pressure mains.

The outlet connections from the vacuum pan are run direct to the

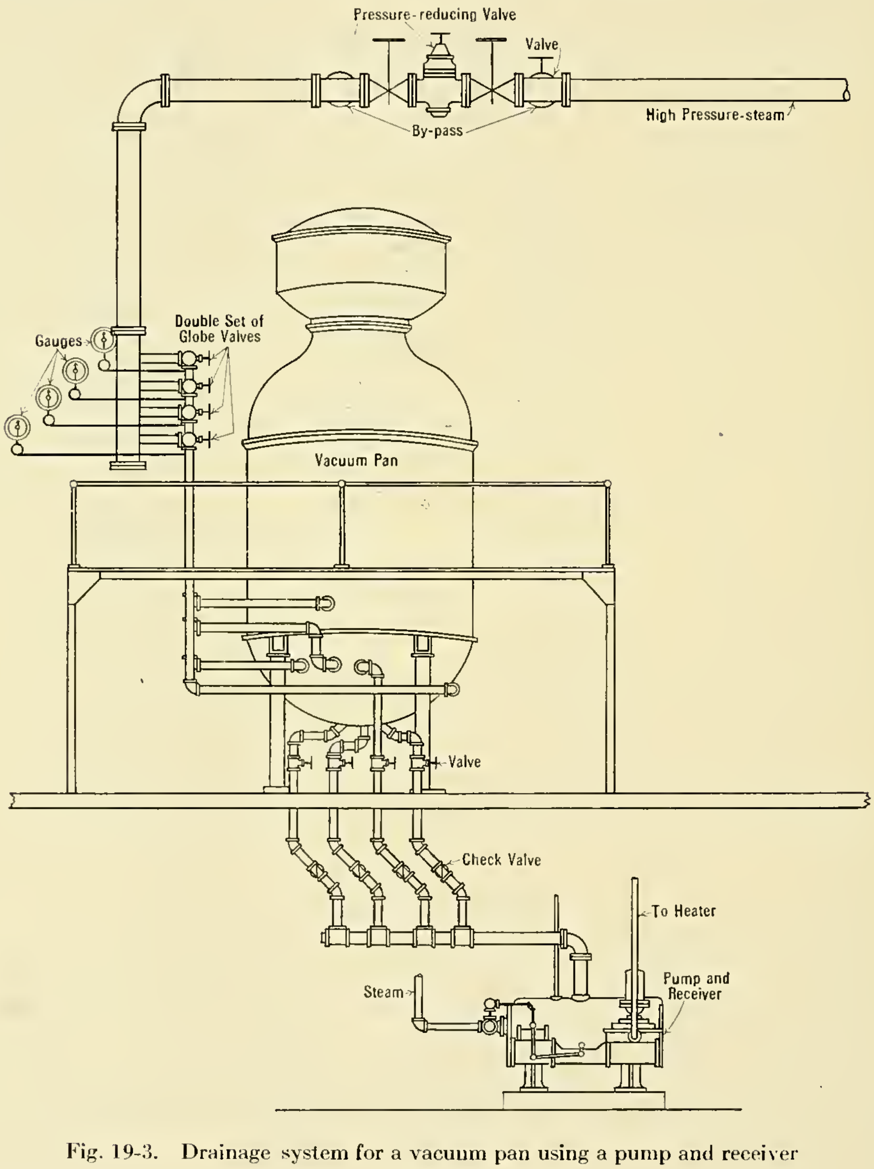


Fig. 19-3. Drainage system for a vacuum pan using a pump and receiver

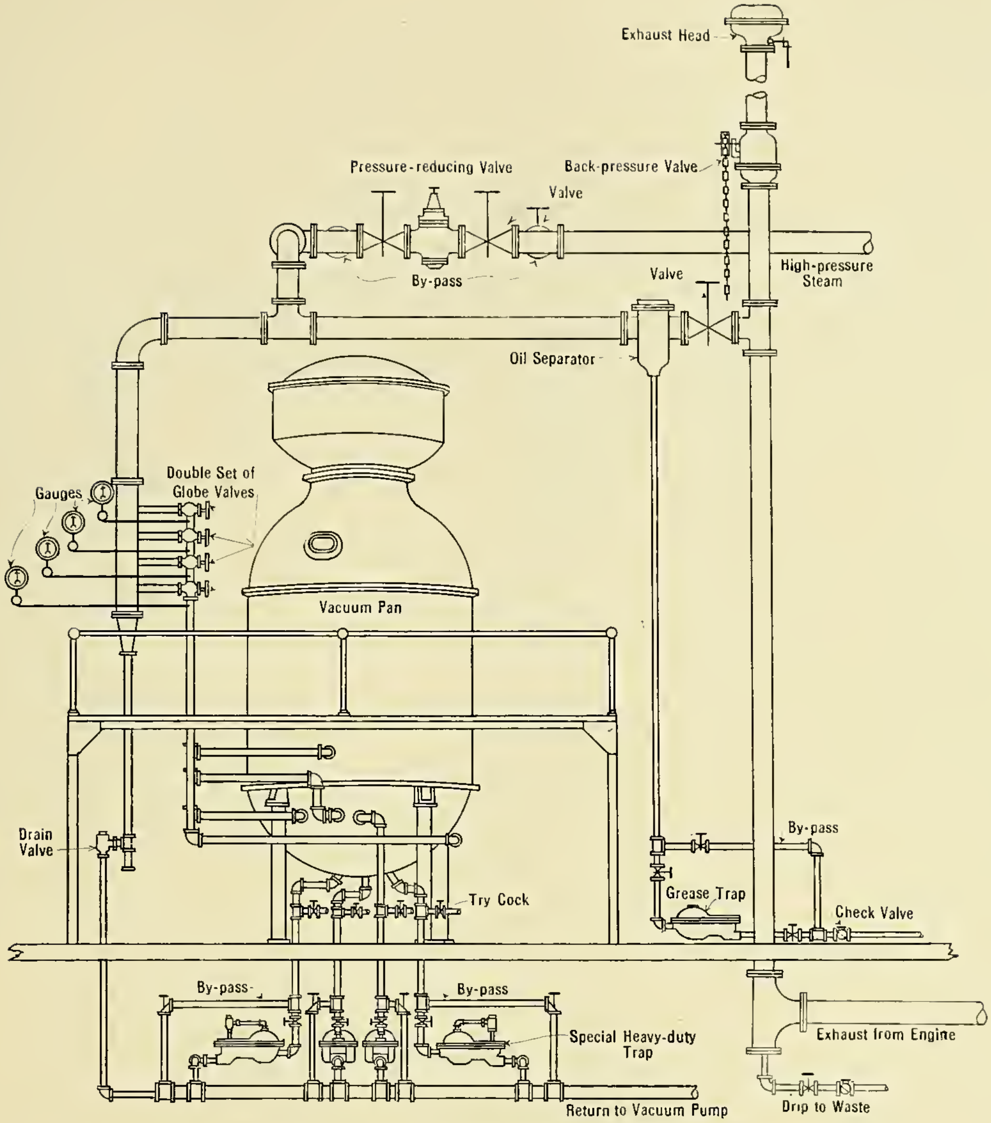


Fig. 19-4. Approved manner of applying the Webster System to a vacuum pan

Webster High-differential Heavy-duty Traps, which are provided with by-passes and thermostatically controlled air lines and are connected directly to the vacuum return line, which is run through a Webster Suction Strainer to the vacuum pump. These outlet connections also must be equipped with small try-cocks in order that the operator may test the working condition of any coil in the pan at any time.

CHAPTER XX

Application of the Webster System to Sterilizers, Cooking Kettles and Similar Apparatus

HOSPITAL EQUIPMENT: All hospital equipment, such as sterilizers for surgical instruments, bandages and dressings, blanket warmers, etc., requires steam at more than the usual heating pressures. As these fixtures are comparatively small consumers of steam, being operated at gauge pressures of 15 to 100 lb., and as they are situated at different parts of the building, it is usual to run a special set of steam supply and return lines for them so that steam may be available at any time throughout the year.

For the purpose of insuring rapid removal of condensation and air from each fixture, a Webster Return Trap of similar construction to those described in the preceding chapter is placed on the return of each unit. The operating temperature of the thermostatic members of these traps is close to that of steam at atmospheric pressure; hence it is necessary to provide sufficient exposed piping between the fixture and the trap to allow the condensation to cool down to the operating temperature of the return trap. This exposed piping is termed cooling surface.

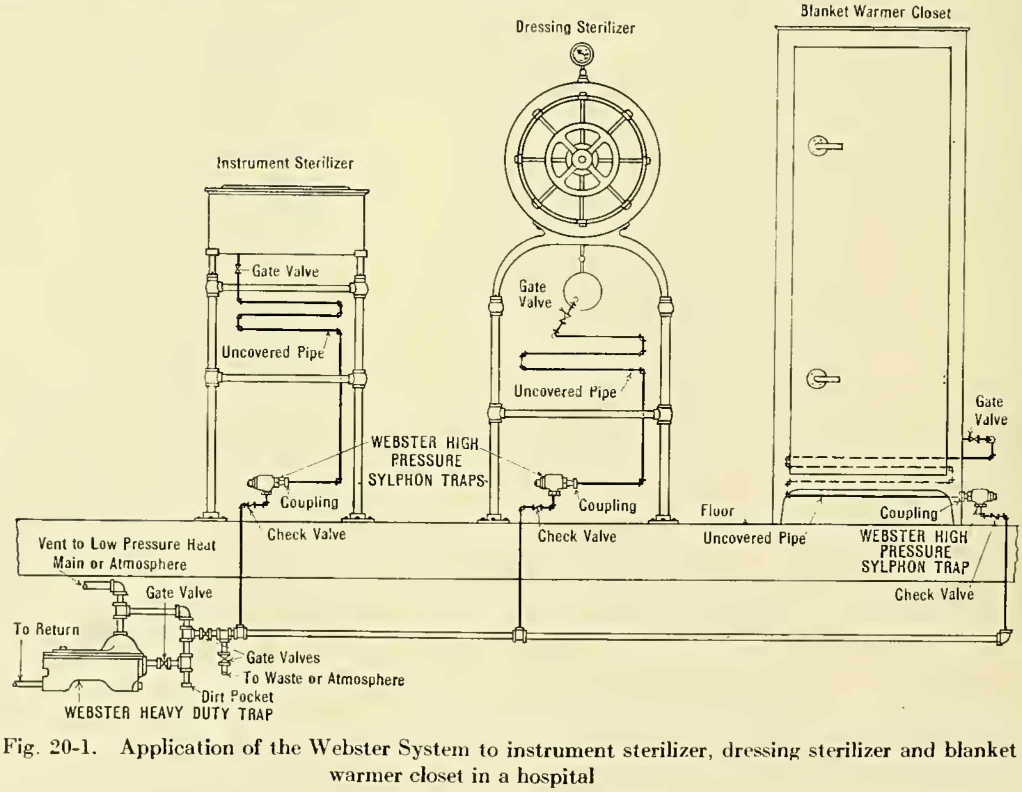


Fig. 20-1. Application of the Webster System to instrument sterilizer, dressing sterilizer and blanket warmer closet in a hospital

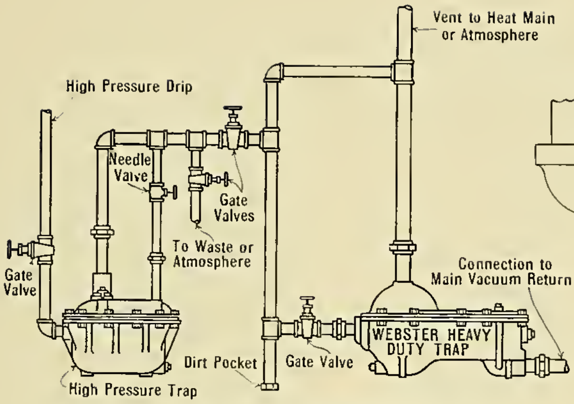


Fig. 20-2. Method of discharging high-pressure drips or returns from high-pressure apparatus into low-pressure heating mains and vacuum return mains through a Webster Heavy-duty Trap

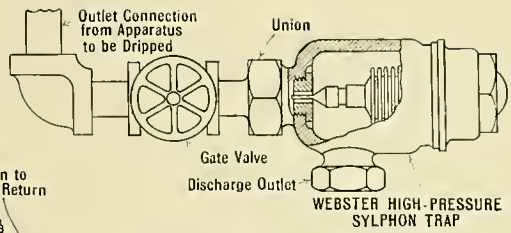


Fig. 20-3. Connections for return trap where the operating pressure exceeds 10 lb. per sq. in.

Each return from trap before connecting into the common discharge line of similar traps should have a check valve between the trap and the return as well as a hand shut-off valve between fixture and trap as shown in Figures 20-3 and 20-4. *This is very important as a protection for the trap against water hammer.*

Where several Webster High-pressure Sylphon Traps discharge their condensation into a common return line, it is necessary that this line shall be vented free to the atmosphere, or in cases where possible, to the low-pressure heat main (Figure 20-2). *It is important that no back pressure shall be carried on this return line.* In no case should the discharge of these traps be connected directly to a vacuum return as the vacuum

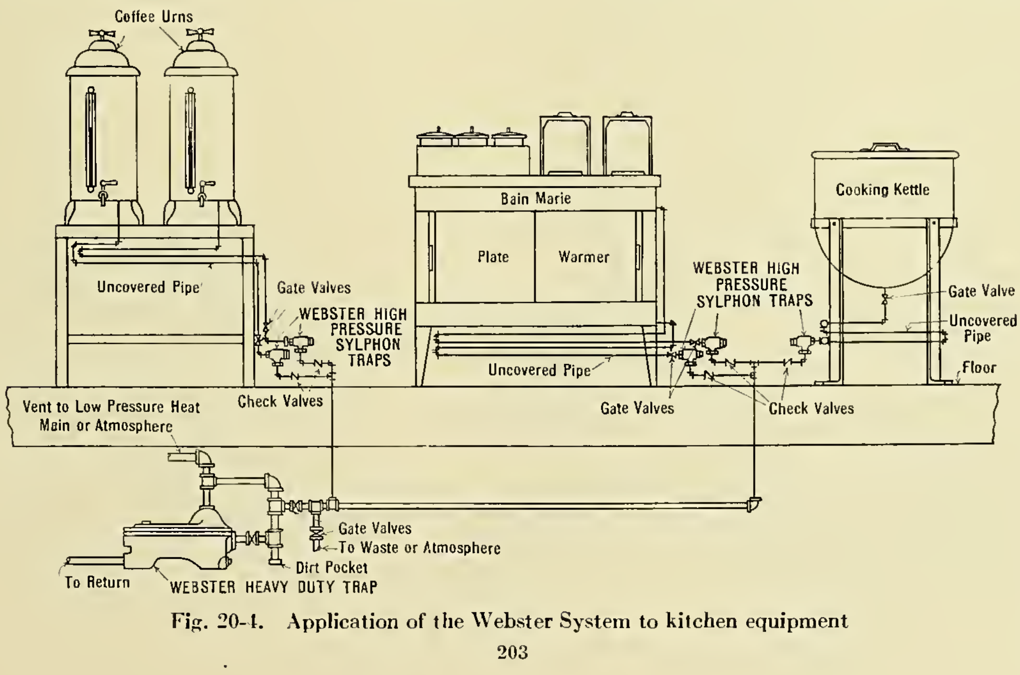


Fig. 20-4. Application of the Webster System to kitchen equipment

would unbalance the operating member of the trap and cause it to give unsatisfactory results.

COOKING KETTLES, PLATE WARMERS, BAIN-MARIES, COFFEE URNS AND OTHER KITCHEN EQUIPMENT: This equipment requires practically the same treatment as that of hospitals, and the same general statement about arrangement of return lines applies.

In food-product factories where the cooking equipment is much more extensive, a special form of float-controlled return trap with thermostatic trap in air line is used. This particular type is called the Webster High-differential Heavy-duty Trap. For details of these traps see Chapter 24, page 249. These traps are also used for removing the condensation and air from the steam coils of vacuum pans in evaporating processes for sugar, milk, salt, tartaric acid, candy, and the like.

It is important in all applications to high-pressure duty that the maximum initial steam pressure to which the trap may be subjected does not exceed the allowable pressure of that class (see Page 275), and that the maximum condensation rate shall be known. It is also important to know in advance the lowest pressure to which the vent of the Heavy-duty Trap will be subjected at times, as the influence of this pressure is marked in limiting the rating of the Webster High-pressure Sylphon Trap.

CHAPTER XXI

Applications of Webster Systems to Greenhouses

THE heating of greenhouses is a special field, owing to the peculiar characteristics of the buildings and the necessity for uniform interior temperatures.

Commercial greenhouses are more exacting in their heat requirements than are public or private conservatories. Constant maintenance of the most desirable temperatures is essential in commercial houses to bring the crop to salable maturity in the shortest possible time and to keep the quantity of first-class product at a maximum throughout the season. A single serious temperature drop for a comparatively short interval may stunt the crop beyond recovery to normal condition within a month's time, and even slight temperature variation renders some kinds of plants more susceptible to certain destructive fungi.

The heat regulation should be flexible to such extent that by applying more or less heat to compensate for loss of sunlight in cloudy weather, the crop can be retarded or forced to reach maturity at the time of the most profitable market. The blossoming of Easter lilies, for instance, requires absolute regulation within a period of a very few days, and failure to meet the time limits results in an almost total loss. The same principle is utilized during the period of uncertain sunshine between November and February to keep the daily production of the majority of varieties of cut flowers more uniform.

Fig. 21-1. Conservatory of the Missouri Botanical Gardens

Owing to the high rate of heat transmission through the glass of which greenhouse enclosures are constructed, the heating system must be capable of quick response to the demands for extra heat during nights, cloudy and cold days, and particularly when a sudden cold wind springs up. Co-operating with the ventilators, the heating system must respond quickly to the demand for less artificial heat, when the heat from the sun's rays tends to increase the interior temperature beyond the point desired.

Until a few years ago, hot water was considered the best medium for circulation in the heating coils of greenhouses. However, as the size and importance of greenhouses have increased, a medium with quicker response in heat flow has become necessary to better meet the many changes in outside temperature and direction and velocity of wind. Steam has proved ideal for this work where the conditions of the individual problem have been carefully analyzed and a suitable heating layout has been applied.

In different types of greenhouses the arrangement of the heating coils varies to suit the particular plants or vegetables grown and to meet the needs of forcing, propagation, etc.

The conservatory group of the Missouri Botanical Garden at St. Louis, Mo., consisting of palm, economic, cycad, succulent and fern houses (Figures 21-1 to 21-5), is heated by the Webster Vacuum System of Steam Heating. These greenhouses are part of the 125-acre botanical garden presented to the public

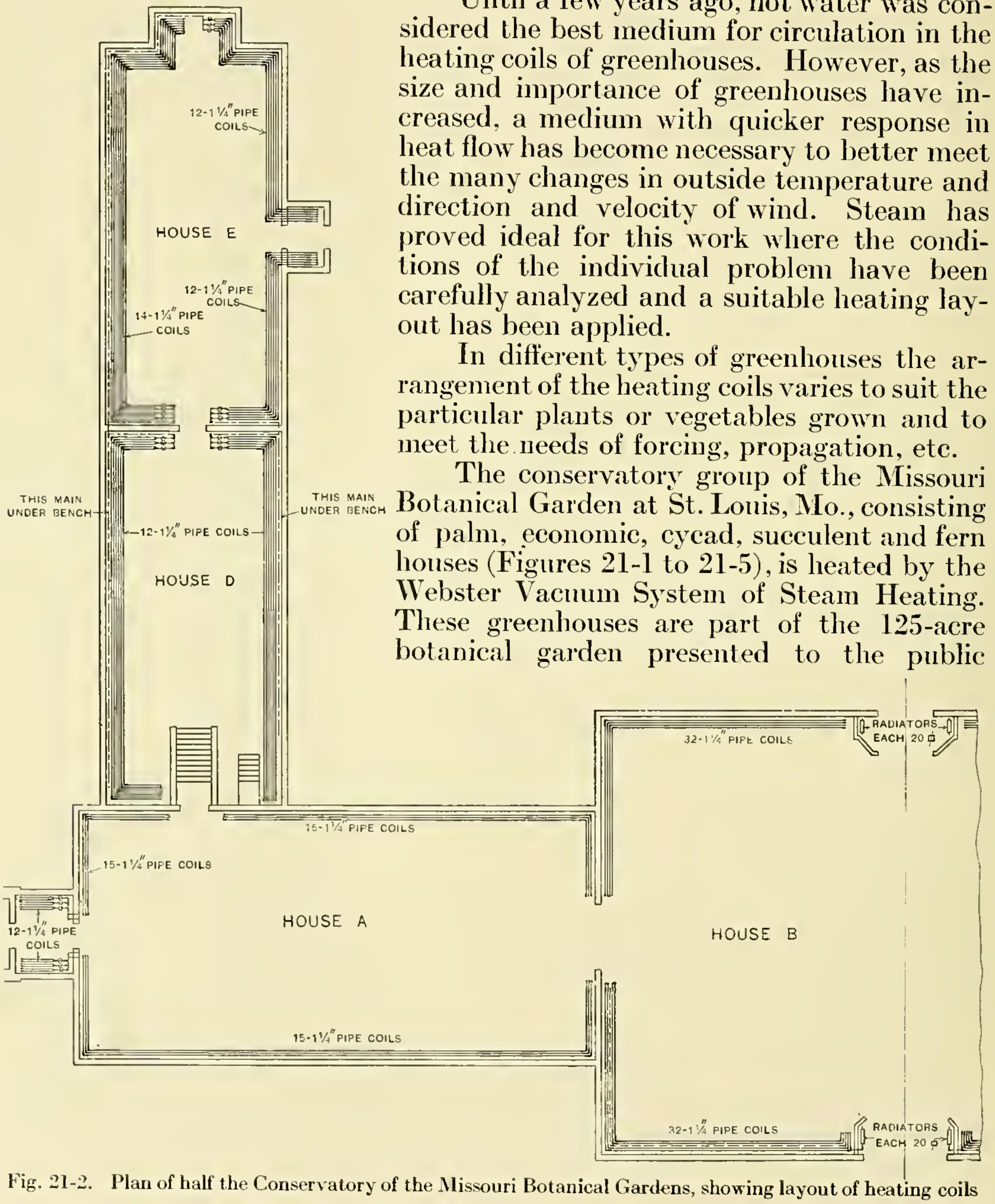


Fig. 21-2. Plan of half the Conservatory of the Missouri Botanical Gardens, showing layout of heating coils

by Mr. Henry Shaw at his death in 1889.

Eleven thousand species of plants grow in this garden. The palm house contains 150 kinds of palms, such as date, cocoanut, sugar, Panama and rattan. The economic house has a variety of tropical and sub-tropical plants, such as rubber, spices, drugs, dyes and coffee. The cycad house is arranged in Japanese style and contains representatives of all known

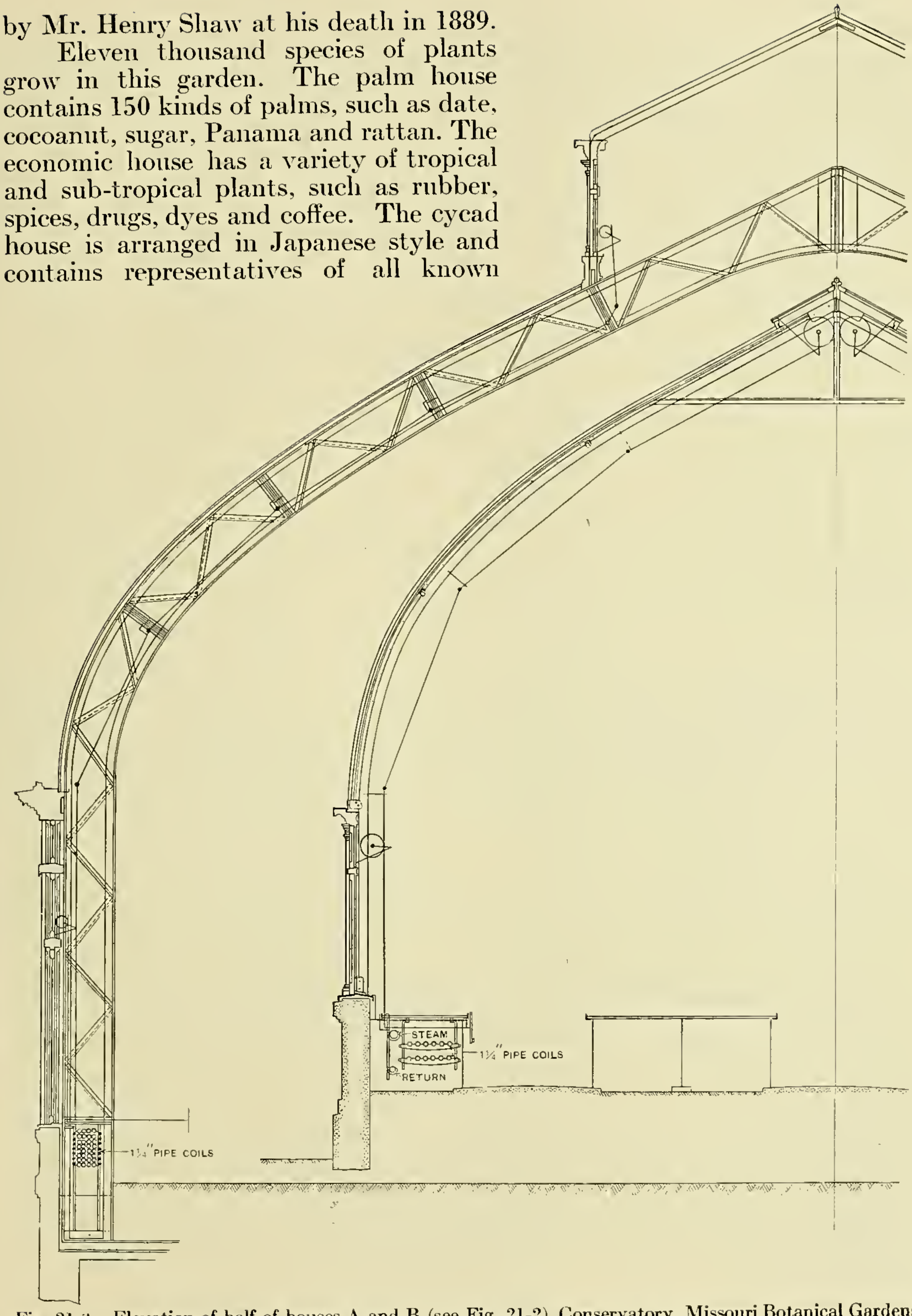


Fig. 21-3. Elevation of half of houses A and B (see Fig. 21-2), Conservatory, Missouri Botanical Gardens. Other halves of these houses are symmetrical with the parts shown

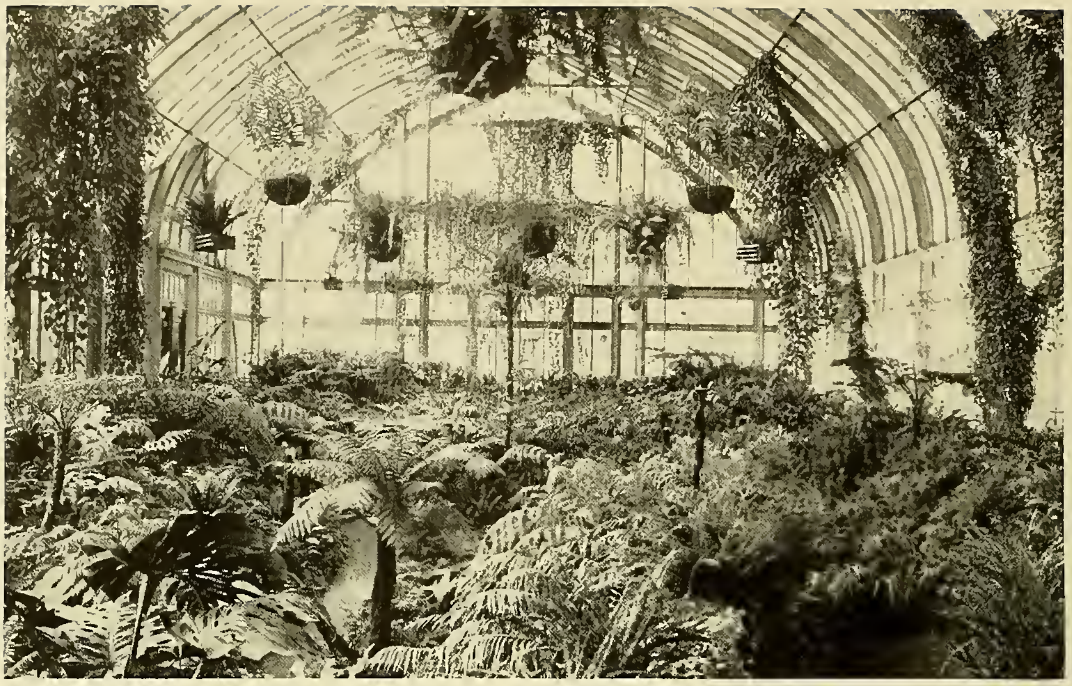


Fig. 21-4. Fern House of the Missouri Botanical Gardens

Fig. 21-5. Floral Display House of the Missouri Botanical Gardens during chrysanthemum show
The accurate temperature regulation obtainable with the Webster System greatly lengthens the prime life of the individual blossoms, thereby assisting in prolonging the duration of the show

genera of cycads, as well as a collection of tropical evergreens. The succulent house contains species of all the plants found in the deserts of the world. The fern house has a very complete collection of the numerous ferns and their allies.

Different atmospheric conditions are required in each of these houses. Ferns, for instance, would not live in the dry air needed by the cacti. The Webster System is maintaining the required temperatures throughout every part of these conservatories, and in most locations the permissible variation in temperature is limited to five degrees.

The palm house is 60 ft. high. To assure maintenance of temperature within 5 deg. variation, the sizing, locating and controlling of radiating surfaces were specially important problems of the design.

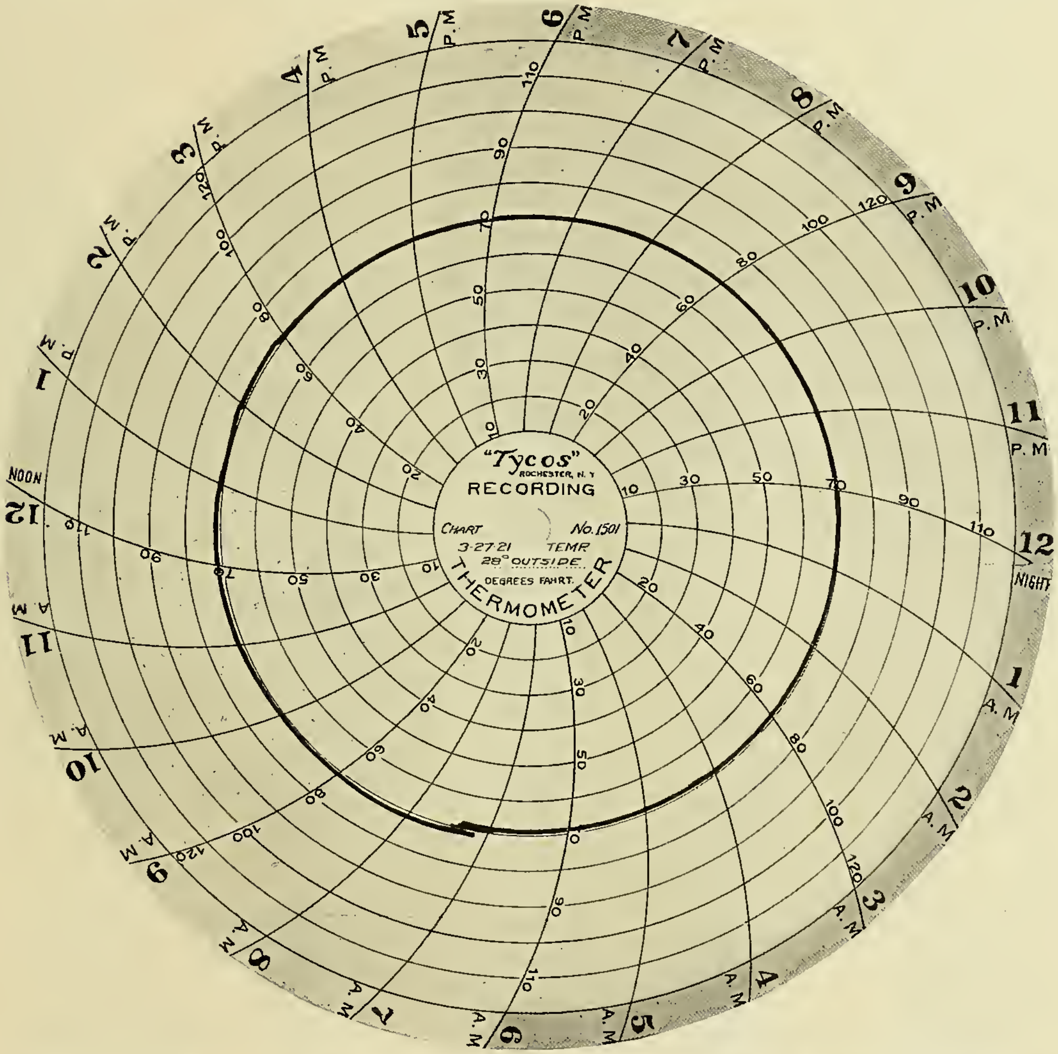


Fig. 21-6. Typical temperature chart from one of the greenhouses of the Davis Gardens, Terre Haute, Ind. The outside temperature on the day the chart was taken averaged 28 deg. fahr. The variation in inside temperature was less than 3 deg. in 24 hours

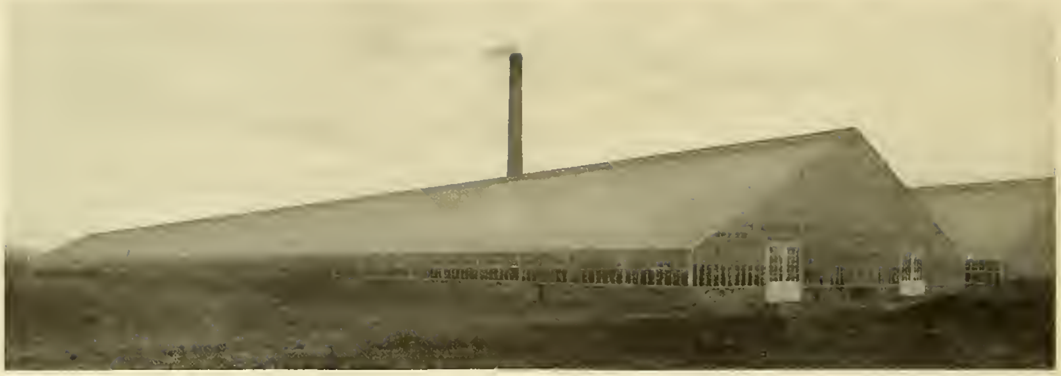


Fig. 21-7. One of the ten 600 by 80-ft. greenhouses of the Davis Gardens, Terre Haute, Ind. In the trade, this establishment is looked upon as a leader in quality of product as well as capacity. The ability to force or retard the crop in each greenhouse assists materially in regulating the output to best meet demand, and in this respect the Webster Vacuum System plays an important part.

Fig. 21-8. Crosswise view at the center of one of the cucumber houses of the Davis Gardens, showing arrangement of heating coils around the beds

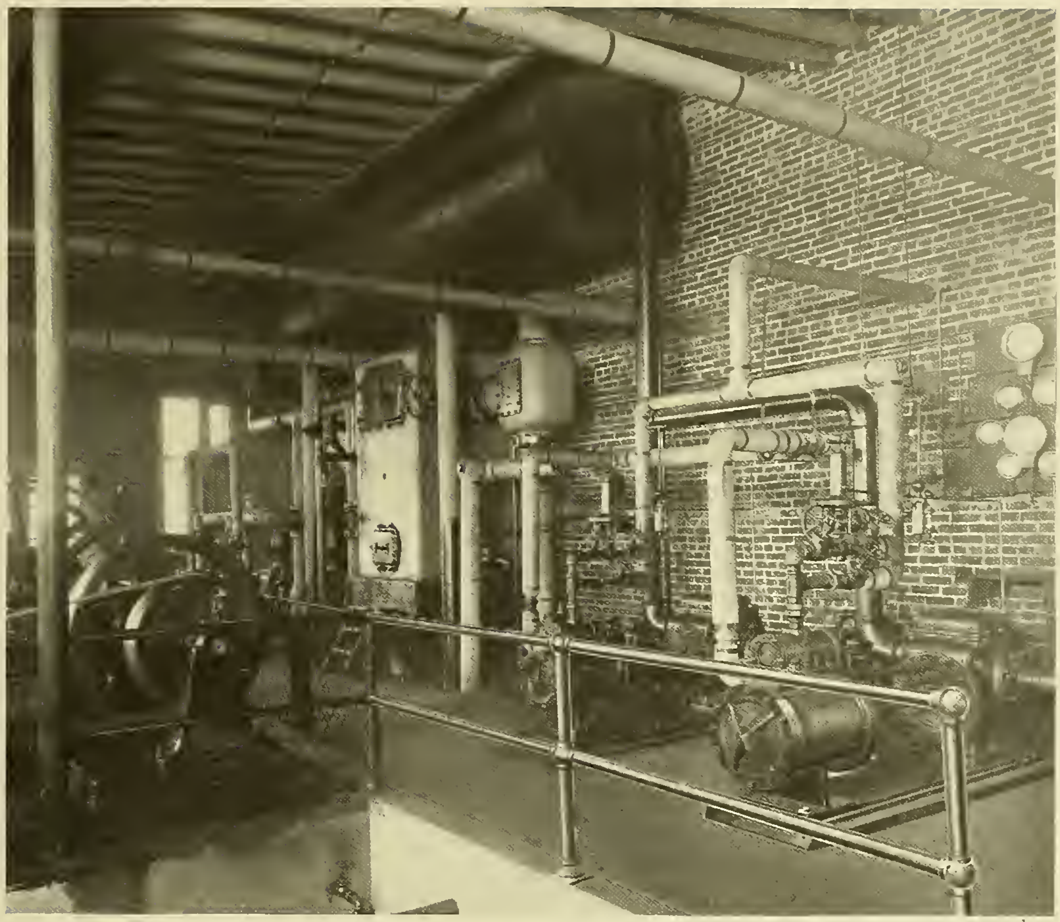


Fig. 21-9. Part of the power plant of the Davis Gardens, showing the feed-water heater and vacuum pumps of the Webster Heating System

The heating coils are banked on the side walls of the houses as shown in Figure 21-3, and the arrangement of the coils is shown in plan, Figure 21-2. Steam is supplied from a central heating plant under pressure and is reduced at the conservatory, the heating system operating at from 1 to 2-lb. gauge pressure. The returns flow to the power house, where the main vacuum pumps discharge the condensation to an open tank, from which it is pumped to the boilers.

The J. W. Davis Company of Terre Haute, Indiana, operates the largest hothouse vegetable growing plant in the country, this plant consisting of 10 greenhouses, each 600 ft. long by 80 ft. wide and one greenhouse, 200 ft. long by 20 ft. wide. Some idea of the magnitude of these houses may be obtained from the fact that for heating alone an 1800-hp. steam generating plant and 60 miles of coils and piping are required.

The main vegetables grown by the Davis Company are hothouse-grown cucumbers, tomatoes and mushrooms. The average output is as follows: cucumbers, 12000 dozen per week; tomatoes, 40000 pounds per

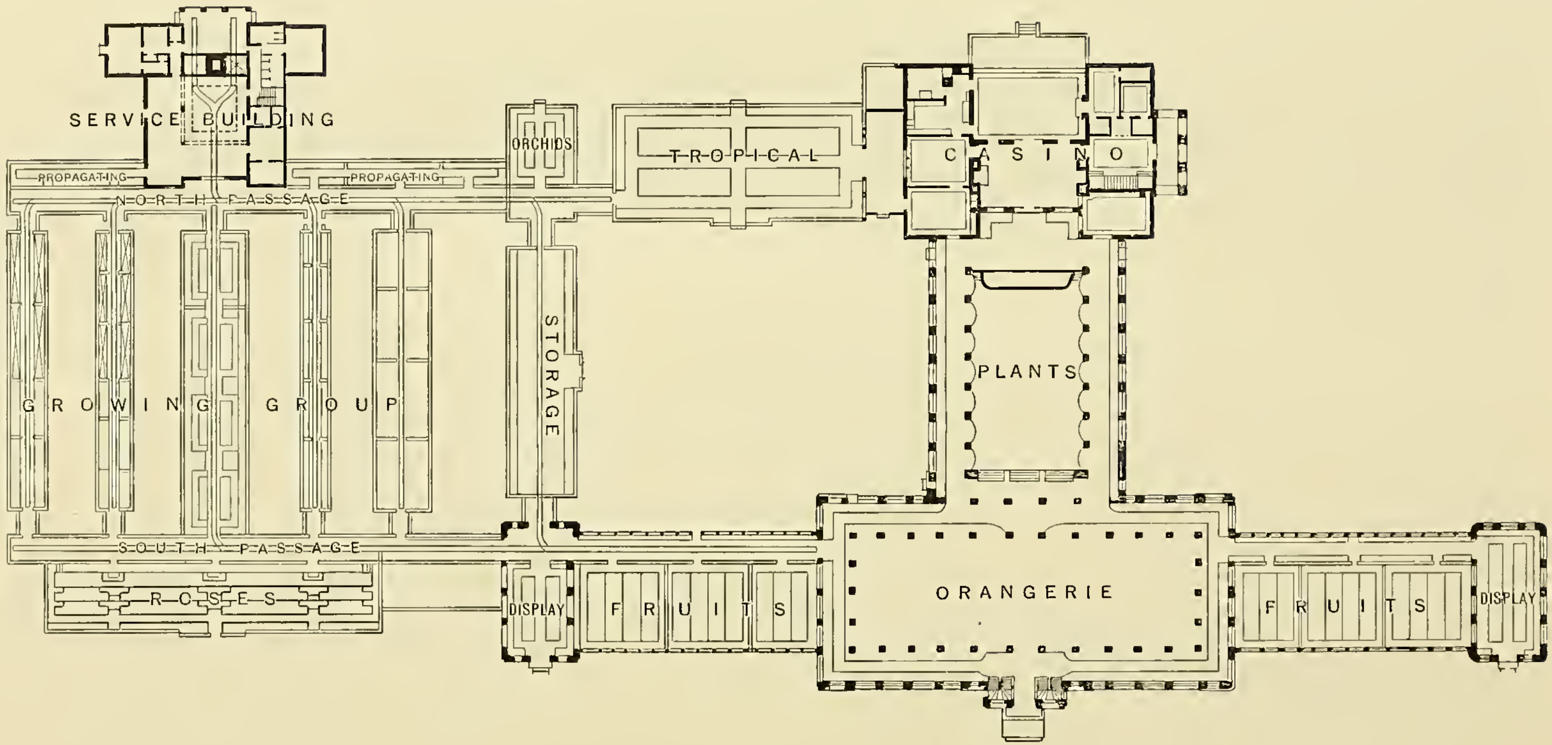


Fig. 21-10. The Horticultural Group on the estate of Pierre S. du Pont, Mendenhall, Pa. Robert P. Schoenijahn, Designing Engineer, Wilmington, Del.

Fig. 21–11. View across Orangerie, du Pont Horticultural Group, Mendenhall, Pa.

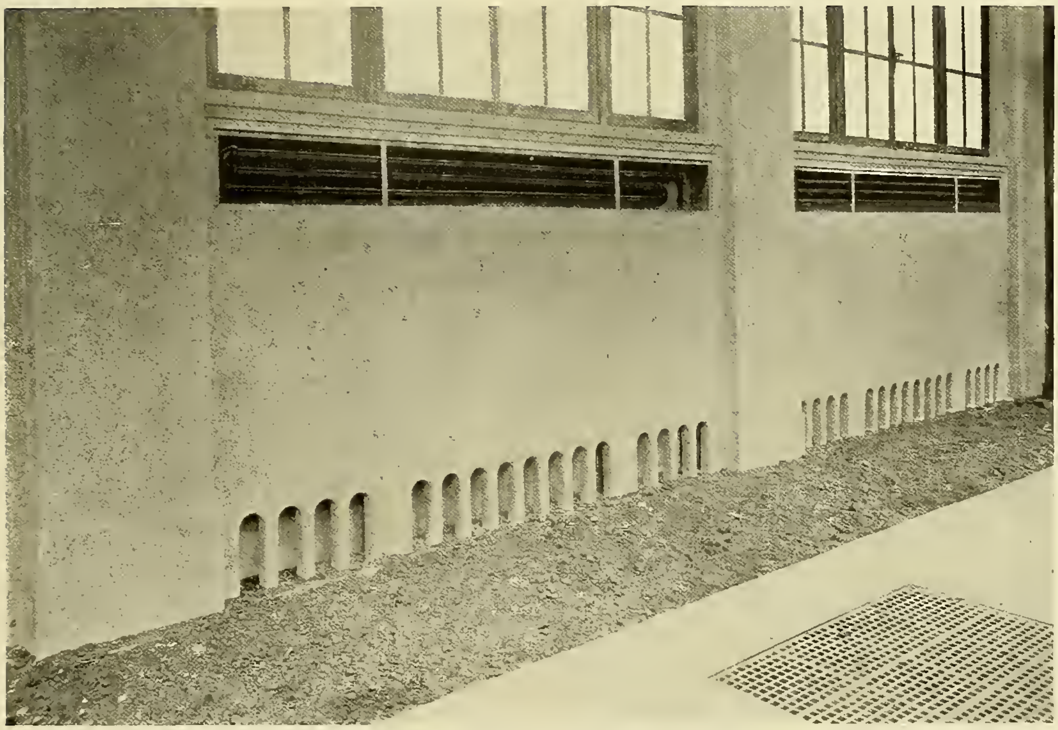


Fig. 21–12. Method of heating for growing vines on the walls of the du Pont orangerie. Air enters the openings at the bottom of the wall, is heated in passing over the coils at the top and passes into the rooms. The registers in the floor distribute heated air from the indirect heating system

week; mushrooms, 2000 pounds per week. The output includes also flowering plants, among which are hundreds of thousands of cyclamen, grown for the sale of cut flowers as well as the plants themselves.

The temperature requirements of these greenhouses are even more exacting than those of the Missouri Botanical Garden, as shown by the chart, Figure 21-6, taken from the recording thermometer.

The steam for heating is taken from a 95-lb. steam line running through the connecting corridors, and the pressure is reduced in each greenhouse for the Webster Vacuum Heating System, which operates at 5-lb. pressure. The condensation is carried through a vacuum return back to the power plant, where it is delivered by the main vacuum pumps through a tank to a Webster Feed-water Heater and from there pumped to the boilers.

The Horticultural Group (Figure 21-10) on the private estate of Mr. Pierre S. du Pont near Mendenhall Pa., is heated by the Webster System.

The main buildings comprise the orangerie, exhibition hall, peach houses and display houses. The orangerie is approximately 80 by 180 ft. and the exhibition hall is about 80 by 110 ft. The two peach houses lie on either side of the orangerie and are approximately 50 by 100 ft. in length, with the display house 30 by 50 ft. at the extreme ends. At the rear of the Exhibition Hall is a stage, or rather a veranda, to the future building, which will eventually be the casino. At this end of the building are located the organ and service rooms for entertaining purposes.

The heating for this group is remarkable in that the main buildings are heated by a system of indirect radiation with a gravity circulation of air. The indirect surfaces enclosed with copper casings and pans, are placed in a series of tunnels which lie under the walk-ways. Fresh air when required is taken through two sets of primary heaters located in the orangerie and one set of primary heaters for each of the peach and display-house wings. These are furnished with sufficient surface to maintain the air in the tunnels at 60 deg. fahr.

CHAPTER XXII

Installation Details

MANY of the methods of pipe connections which have been developed by Warren Webster & Company during the past 34 years, and have become standard practice, are shown in this chapter and elsewhere in connection with descriptions of specific apparatus. Most of the illustrations have been published as Webster Service Details and are familiar to the profession and trade. These drawings, which indicate the general arrangement of the pipe, fittings and Webster apparatus have been revised from time to time and, as shown here, represent the latest and best thought. They are not to be used for exact layouts of piping, as each individual application presents its own special conditions. No effort has been made to indicate the necessary unions or right and left nipples required for the connections, as these requirements for any case would naturally be best determined by the detail of the layout or by the steamfitter at the job, based upon his skill and upon materials available.

Details Applicable to Both the Webster Vacuum System and the Webster Modulation System

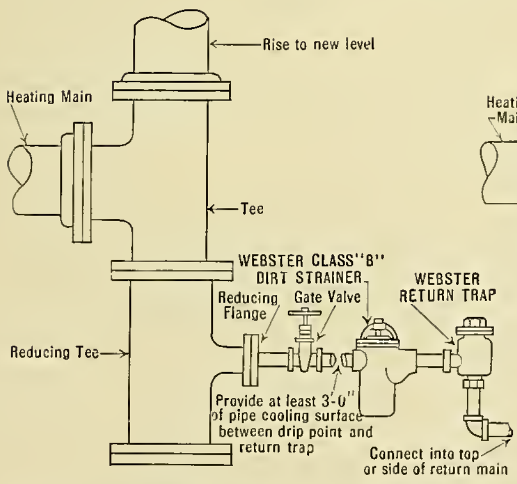


Fig. 22-1. Application of a Webster Return Trap on a low-pressure heat main, at a low point where the main rises. A sufficient length of uncovered pipe must be provided between the drip point and the return trap

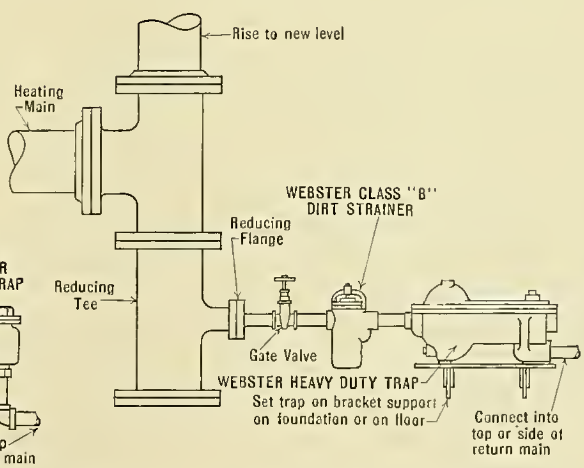


Fig. 22-2. The drainage of a low-pressure heat main at a low point, where the line rises, is of such importance that special attention is warranted. This diagram shows a large main with drip through gate valve, Webster Dirt Strainer and Webster Heavy-duty Trap

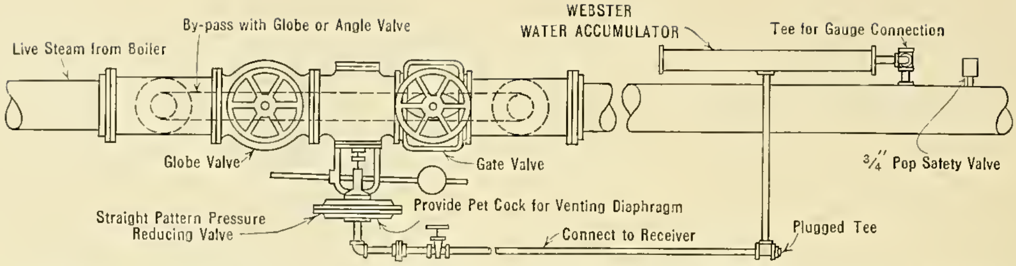


Fig. 22-3. Connections for a steam pressure-reducing valve. The control pipe from the low-pressure side of the line must be taken from a point far enough from the valve to insure that the pressure will have been fully expanded. The use of the Webster Water Accummulator (see Page 267) facilitates a constant static pressure on the diaphragm of the pressure-reducing valve. The pop safety valve prevents pressure building up, particularly at very light loads

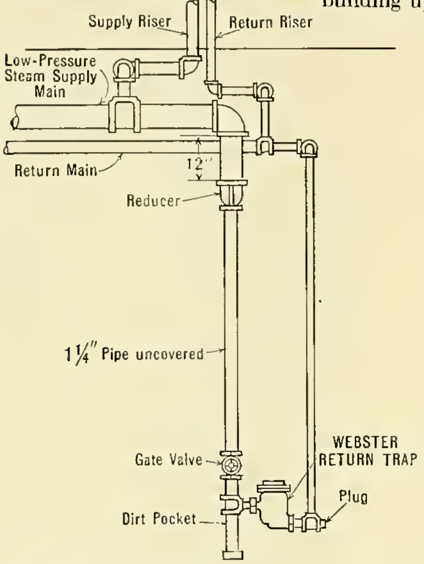


Fig. 22-4. Method of dripping supply risers through a Webster Return Trap into vacuum return line; the vertical leg acts both as cooling surface and dirt pocket

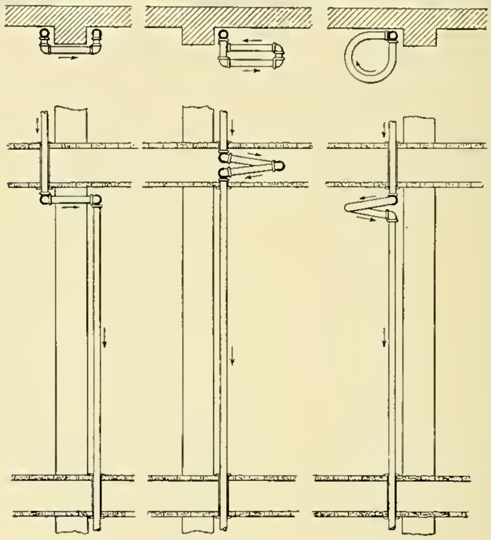


Fig. 22-5. Three methods of making loops to provide for expansion movement in risers. The expansion of supply and return risers should have careful study

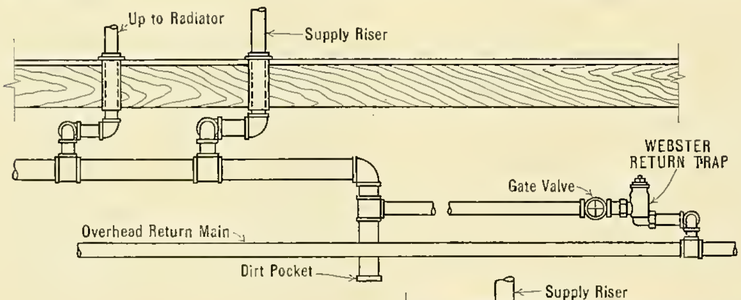


Fig. 22-6. Arrangement for dripping the end of a supply main, which also carries the condensation from the upfeed risers, into an overhead return main. The return trap is located at a point four feet or more from the point dripped

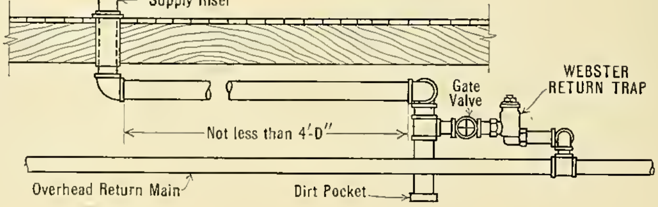


Fig. 22-7. Arrangement for dripping a down-feed riser into an overhead return main, showing the uncovered horizontal cooling pipe

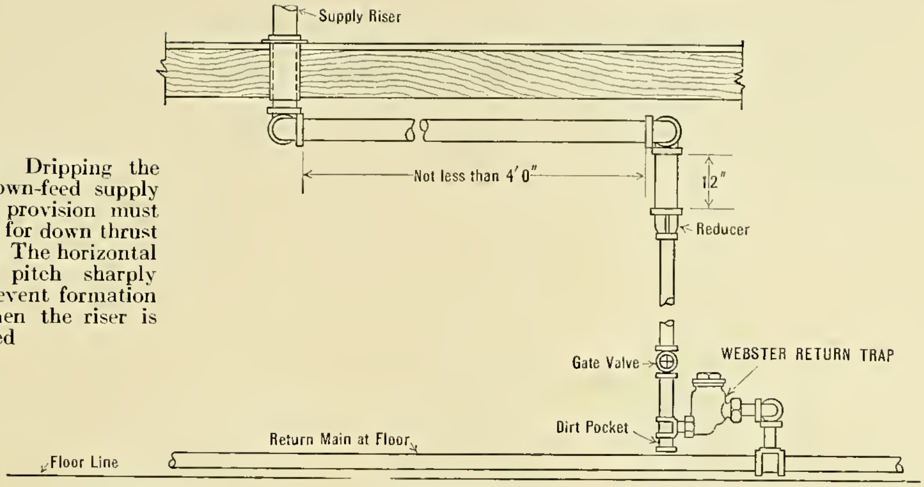


Fig. 22-8. Dripping the heel of a down-feed supply riser, where provision must also be made for down thrust or expansion. The horizontal pipe must pitch sharply enough to prevent formation of pocket when the riser is fully expanded

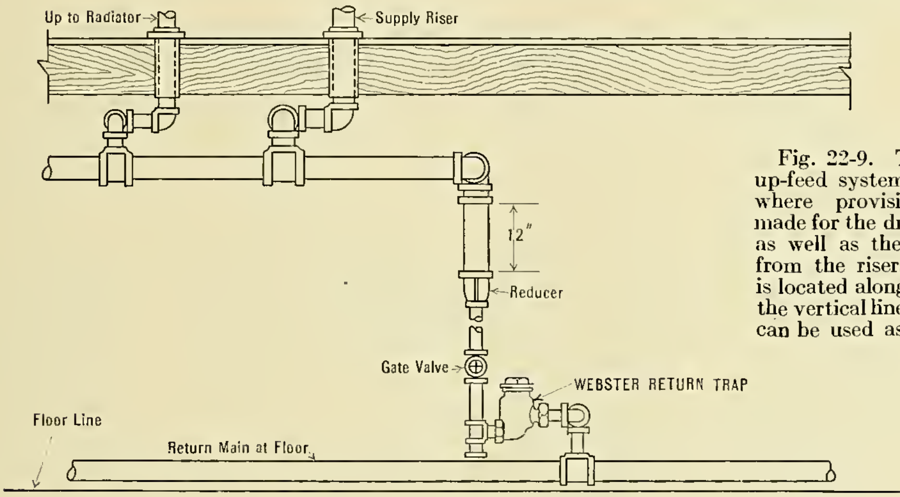


Fig. 22-9. The end of an up-feed system supply main where provision must be made for the drip of the main as well as the condensation from the risers. The return is located along the floor and the vertical line to return trap can be used as a cooling leg

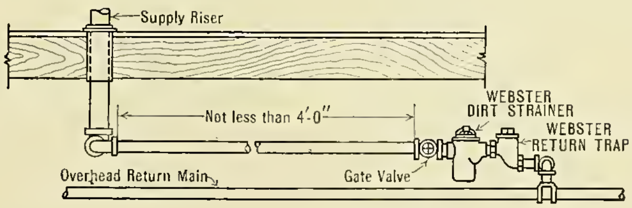


Fig. 22-10. Arrangement for dripping down-feed risers into an overhead return line. Cooling pipe used with a Webster Dirt Strainer located at the entrance to the return trap. The horizontal pipe must pitch sharply downward to prevent formation of pocket

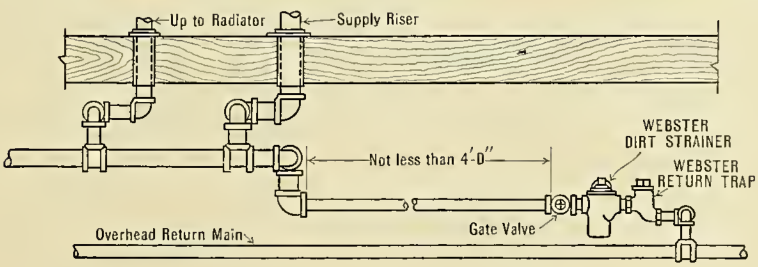


Fig. 22-11. Arrangement for dripping the end of an overhead supply main through Webster Dirt Strainer and Return Trap into an overhead return main

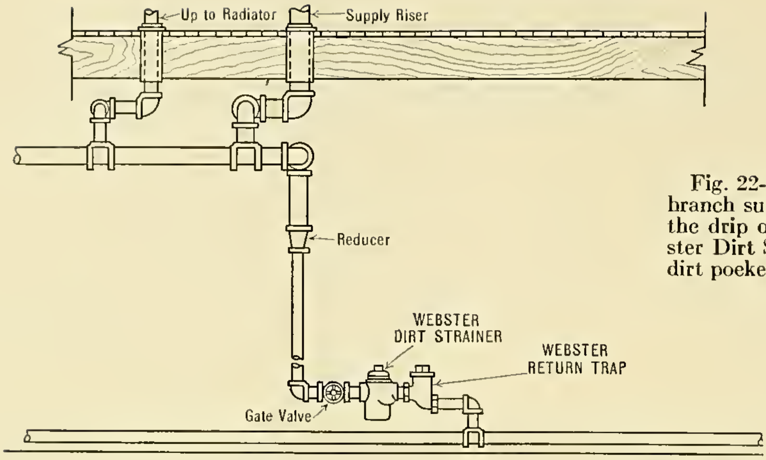


Fig. 22-12. The drip of the end of a branch supply main which also carries the drip of down-feed risers. A Webster Dirt Strainer is used in place of a dirt poeket

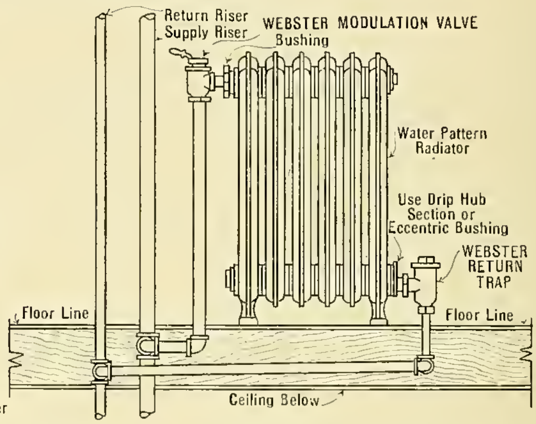


Fig. 22-14. Arrangement of connections to a hot-water type radiator where the branch run-outs are in the floor construction

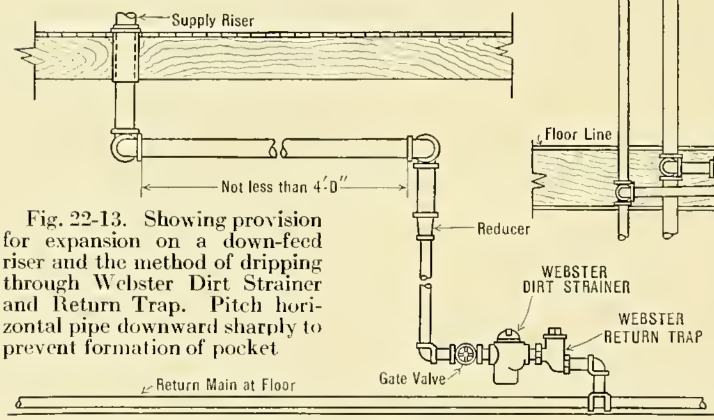


Fig. 22-13. Showing provision for expansion on a down-feed riser and the method of dripping through Webster Dirt Strainer and Return Trap. Pitch horizontal pipe downward sharply to prevent formation of pocket

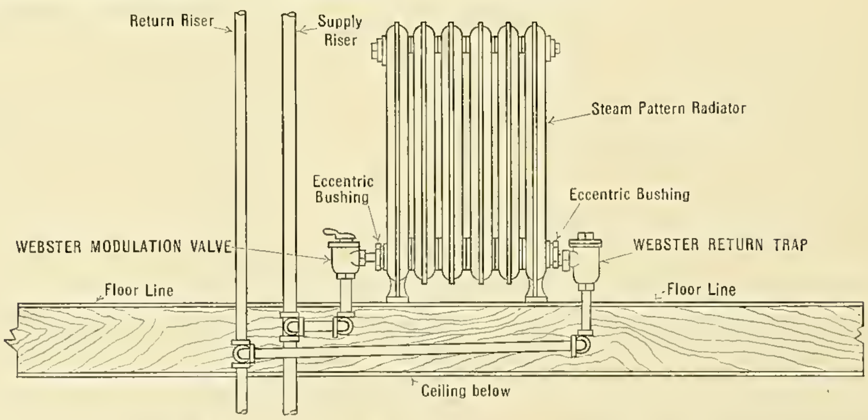


Fig. 22-15. Arrangement of connections to a steam-type radiator where the branch run-outs are in the floor construction

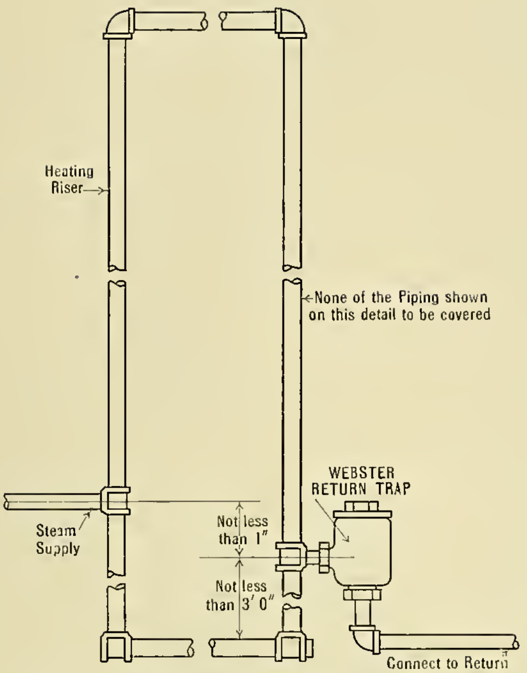


Fig. 22-16. In certain classes of buildings a small amount of heating surface is often desired in bath rooms, etc., without involving the expense of separate radiators. Where these rooms are one above the other a heating riser may be used with connections as shown in this diagram

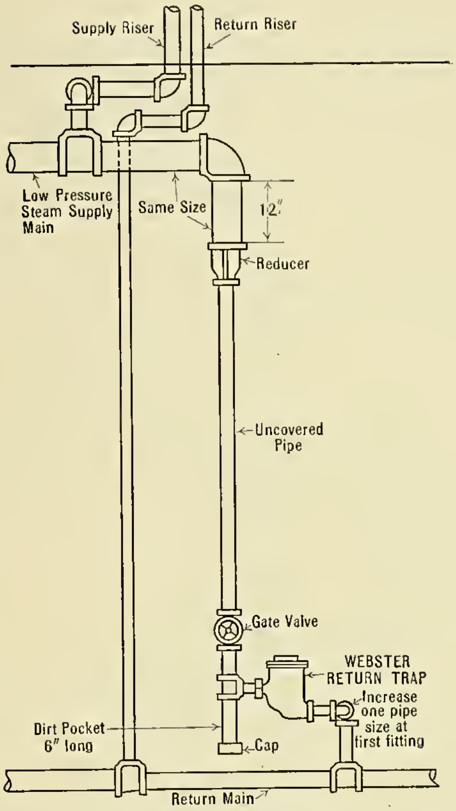


Fig. 22-17. The dripping of the end of an overhead steam supply main where the return line is carried along near the floor. The uncovered vertical line to the return trap acts as a cooling leg

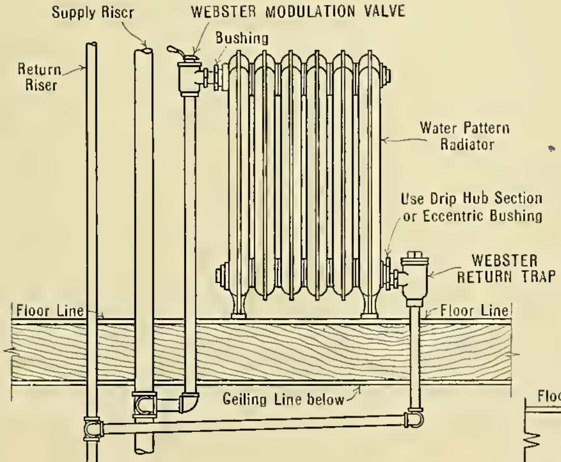


Fig. 22-18. Arrangement of connections to a radiator in a factory or loft building where there is no objection to branch run-outs on the ceiling of the floor below

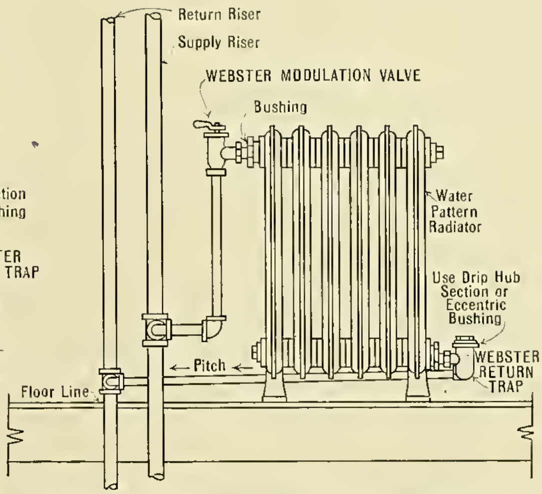


Fig. 22-19. Arrangement of connections to a radiator with all branch run-outs exposed in the room

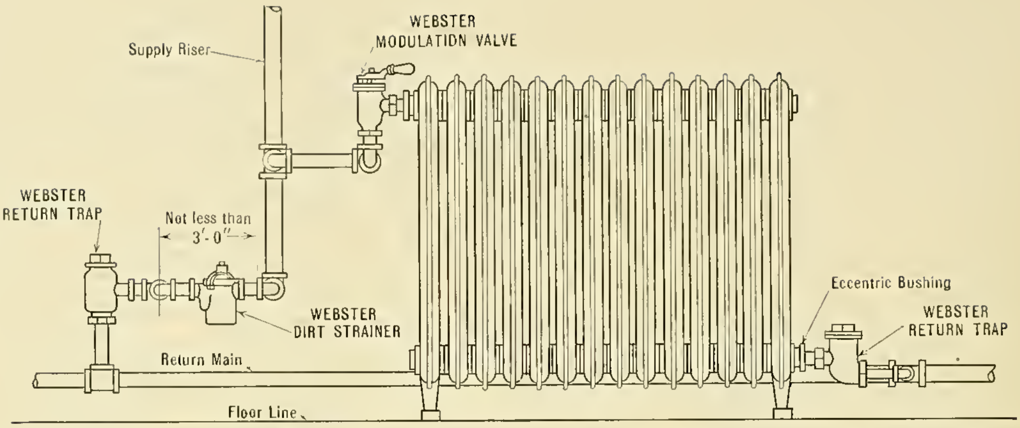


Fig. 22-20. Arrangement for removing a considerable amount of condensation from a down-feed riser. The drip goes through a Webster Dirt Strainer and Return Trap, the connection to lowest radiator being made above the drip point. Fig. 21-23, page 253, shows an alternate method using a Webster Double-service Valve

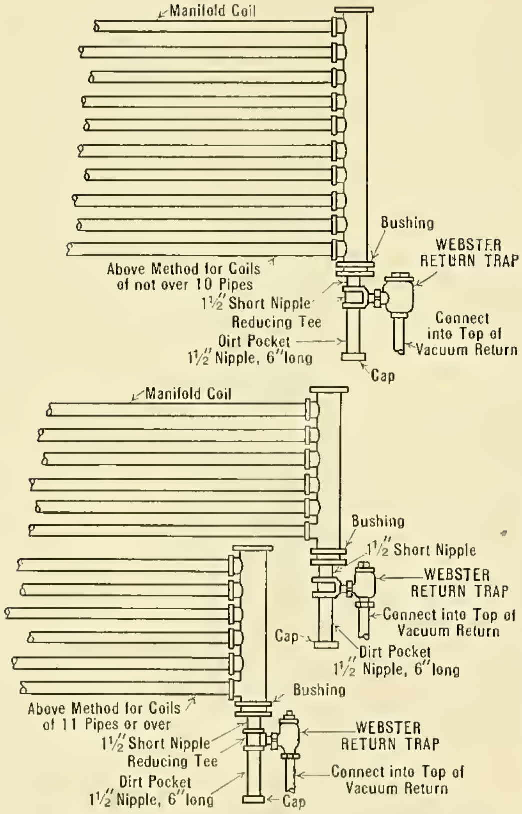


Fig. 22-21. Drip connections to the return headers of manifold coils. Coils of ten pipes or less have one return header and those of over ten pipes are usually split and provided with two headers. Fig. 21-23, page 253, shows an alternate method using a Webster Double-service Valve

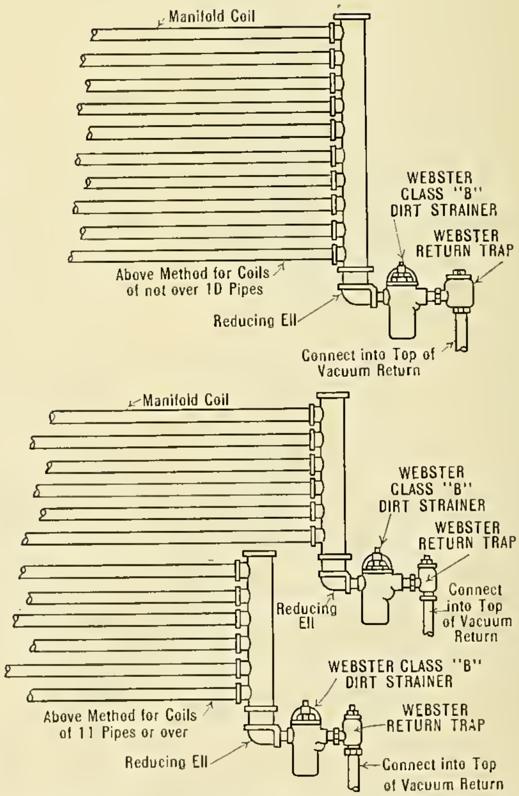


Fig. 22-22. Arrangement of headers similar to Fig. 22-21, but showing the use of the Webster Dirt Strainer at the entrance of the return traps

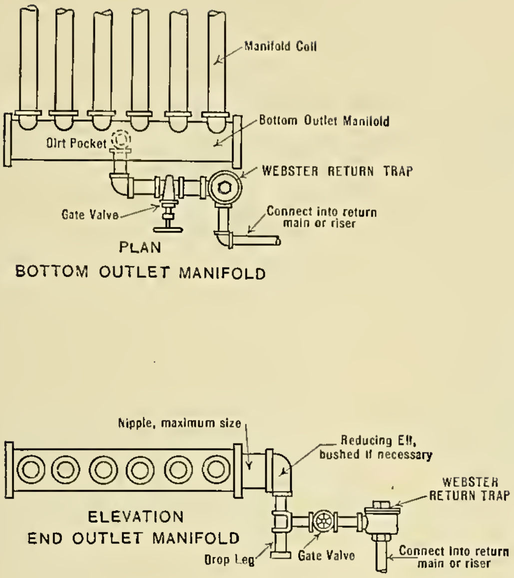


Fig. 22-23. With drop leg to catch dirt

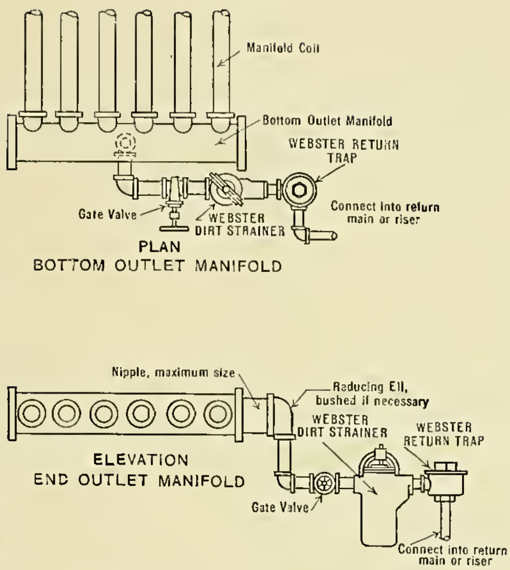


Fig. 22-24. With Webster Dirt Strainer

Return connection to a flat overhead coil where (above) a bottom-outlet manifold and where (below) an end-outlet manifold is used. Dirt is collected by drop leg (Fig. 22-23) or by a Webster Dirt Strainer (Fig. 22-24)

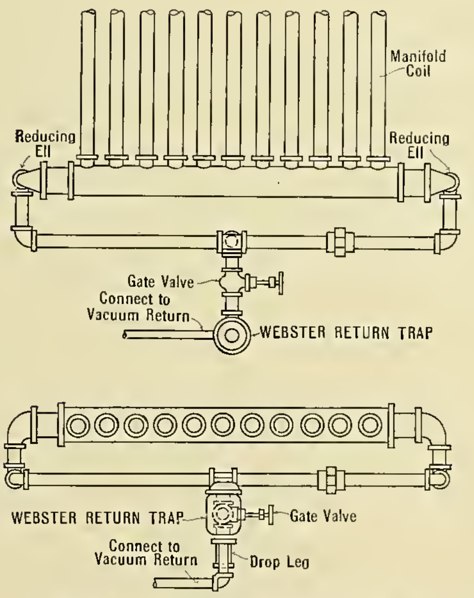


Fig. 22-25. With drop leg to catch dirt

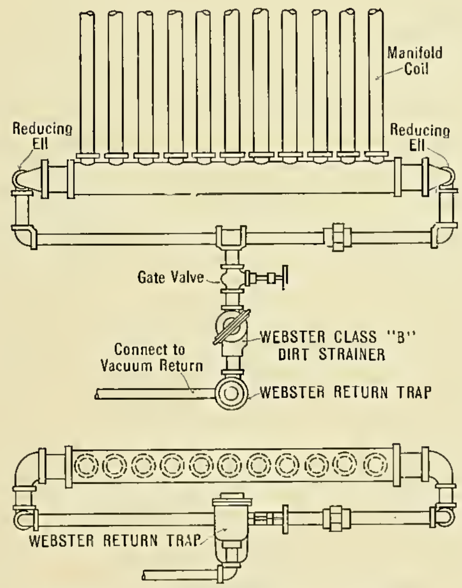
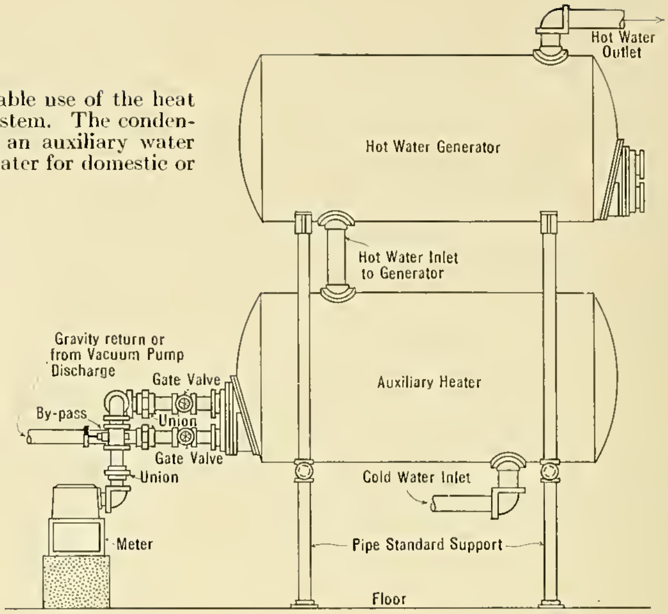


Fig. 22-26. With Webster Dirt Strainer

Wide flat overhead coils should have return connections taken from both ends of the return manifold. Dirt is collected by a drop leg (Fig. 22-25) or by a Webster Dirt Strainer (Fig. 22-26)

Fig. 22-27. Arrangement for profitable use of the heat in the condensation from a heating system. The condensation is passed through the coils of an auxiliary water heater and its heat is transferred to water for domestic or manufacturing use



Note—Additional details applicable to the Webster Modulation System will be found on pages 228 to 232

Details Applicable to the Webster Vacuum System Only

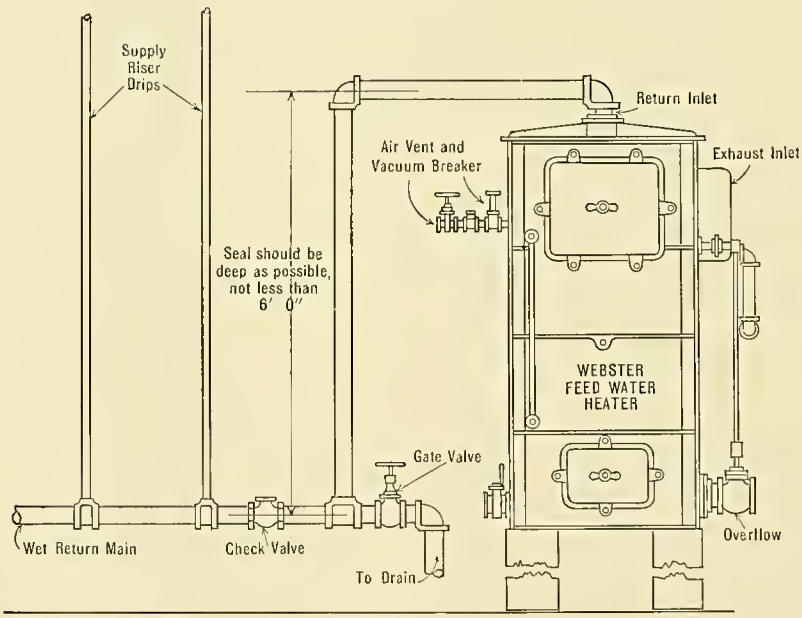


Fig. 22-28. Under certain conditions the condensation from the heels of downfeed risers can be removed by connecting the separate gravity drip or wet-return line to the return inlet of a Webster Feed-water Heater. In this instance, the static head between the top of the heater and the lowest radiator connection must exceed the pressure in the heater. Suitable connection of the return line to the heater is shown in the diagram

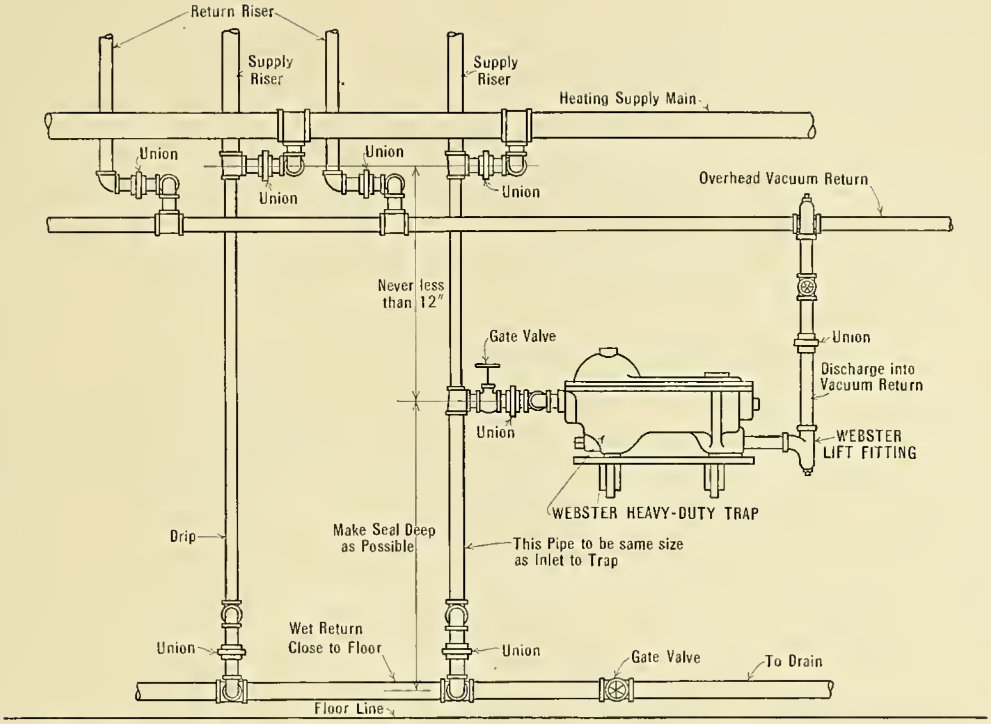


Fig. 22-29. Where the drips of risers and mains are carried through a separate gravity drip line near the floor and it is desired to deliver the condensation into an overhead vacuum return line through a Webster Heavy-duty Trap, the arrangement shown has proved most satisfactory

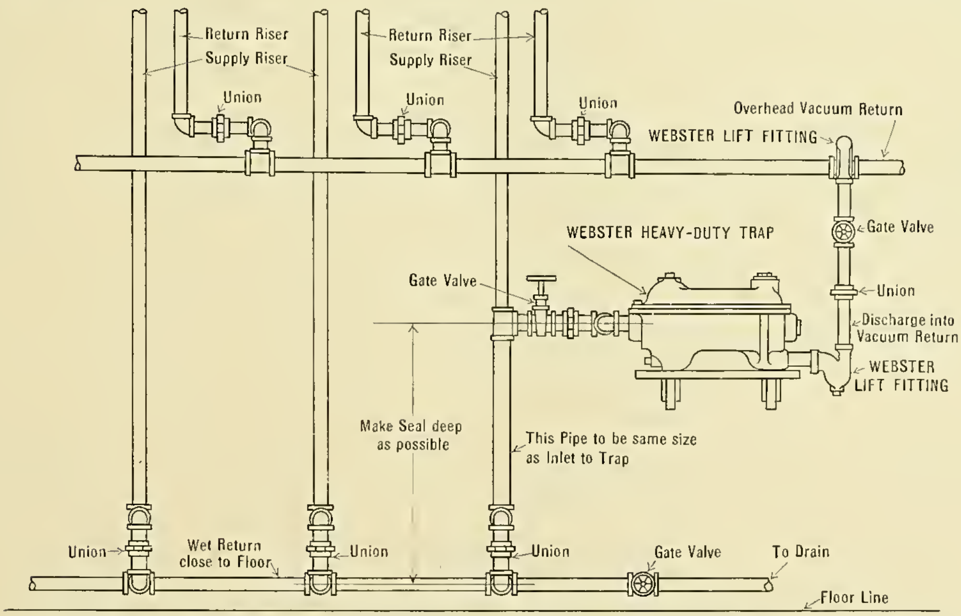


Fig. 22-30. In the usual down-feed system where the drips of risers are cared for by a separate gravity drip line run near the floor and where the condensation is to be delivered to an overhead vacuum return line through a Webster Heavy-duty Trap, the method shown should be followed

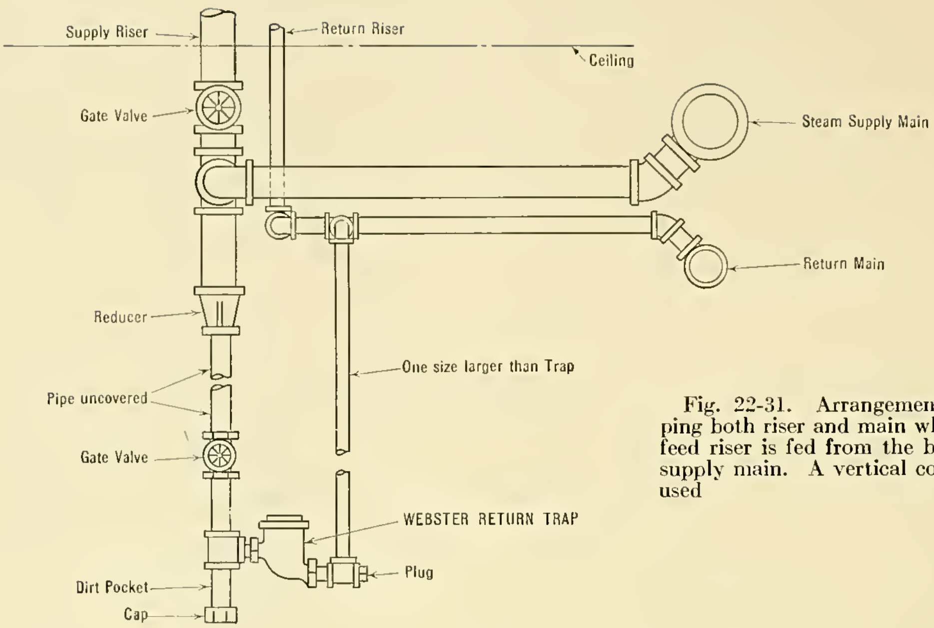


Fig. 22-31. Arrangement for dripping both riser and main where an up-feed riser is fed from the bottom of a supply main. A vertical cooling leg is used

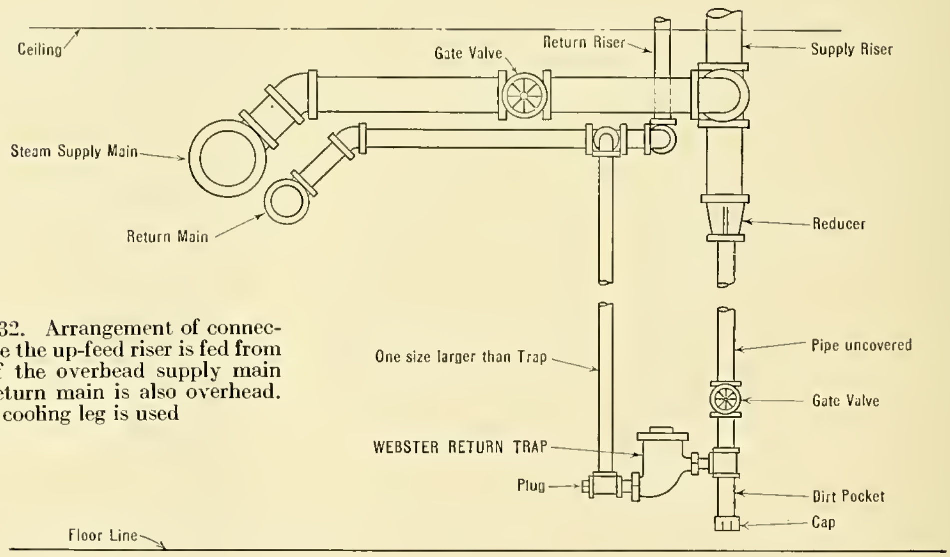


Fig. 22-32. Arrangement of connections where the up-feed riser is fed from the top of the overhead supply main and the return main is also overhead. A vertical cooling leg is used

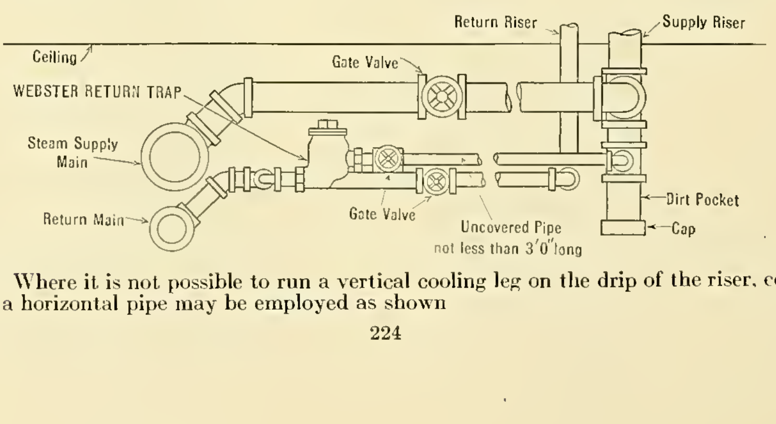


Fig. 22-33. Where it is not possible to run a vertical cooling leg on the drip of the riser, cooling surface in the form of a horizontal pipe may be employed as shown

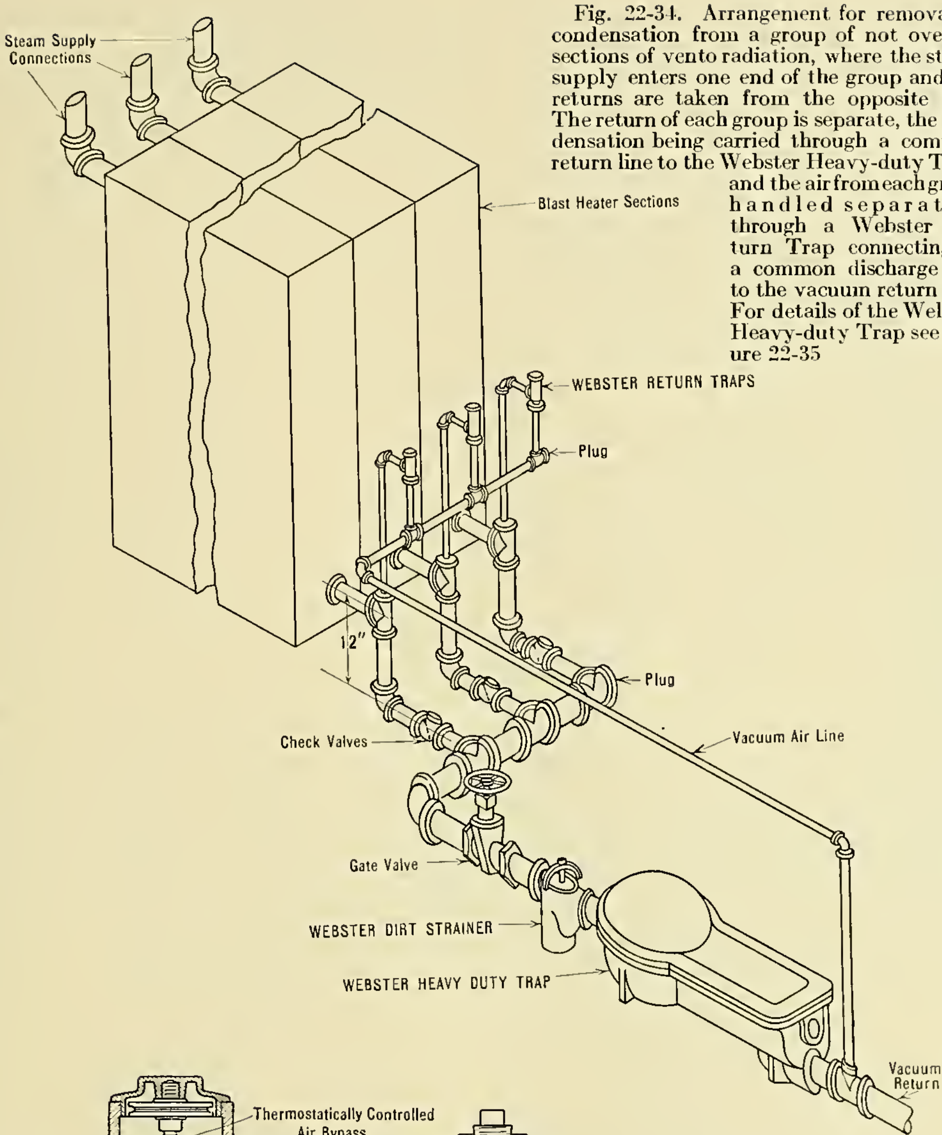


Fig. 22-34. Arrangement for removal of condensation from a group of not over 14 sections of vento radiation, where the steam supply enters one end of the group and the returns are taken from the opposite end. The return of each group is separate, the condensation being carried through a common return line to the Webster Heavy-duty Trap, and the air from each group handled separately through a Webster Return Trap connecting to a common discharge line to the vacuum return line. For details of the Webster Heavy-duty Trap see Figure 22-35

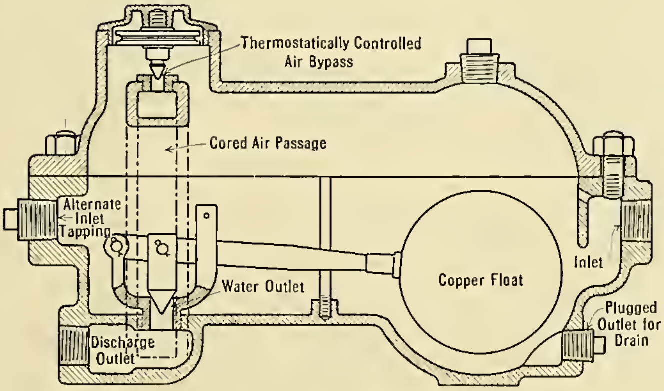


Fig. 22-35. Cross section of Webster Heavy-duty Trap with thermostatically controlled air by-pass, to prevent trap from becoming air-bound

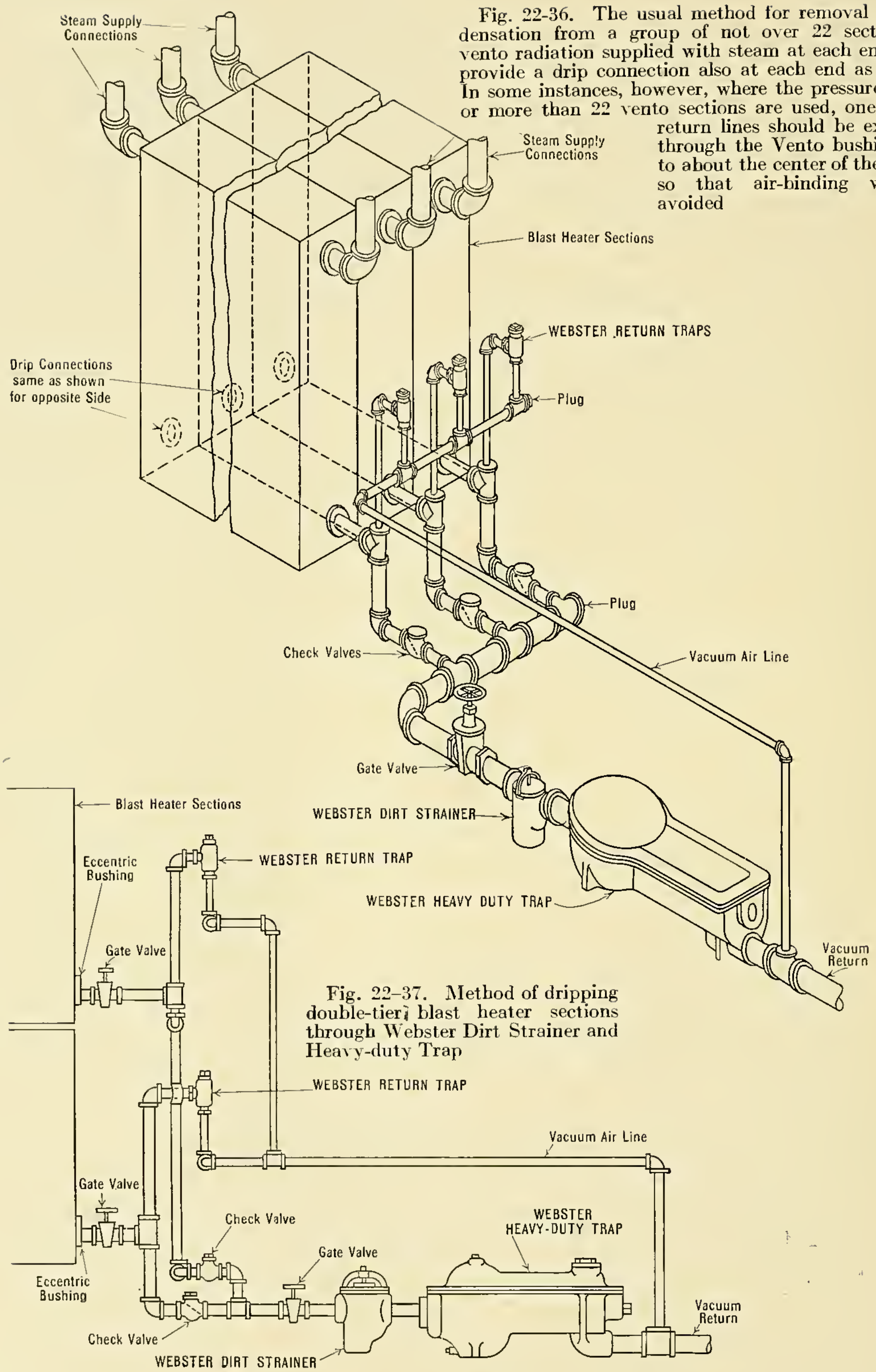


Fig. 22-36. The usual method for removal of condensation from a group of not over 22 sections of vento radiation supplied with steam at each end, is to provide a drip connection also at each end as shown. In some instances, however, where the pressure is low or more than 22 vento sections are used, one of the return lines should be extended through the Vento bushing and to about the center of the group, so that air-binding will be avoided

Fig. 22-37. Method of dripping double-tier blast heater sections through Webster Dirt Strainer and Heavy-duty Trap

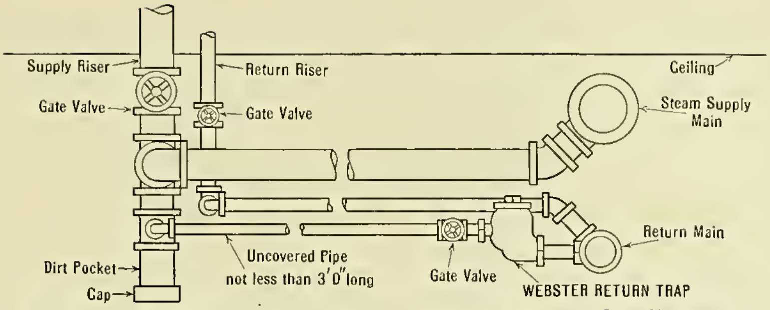
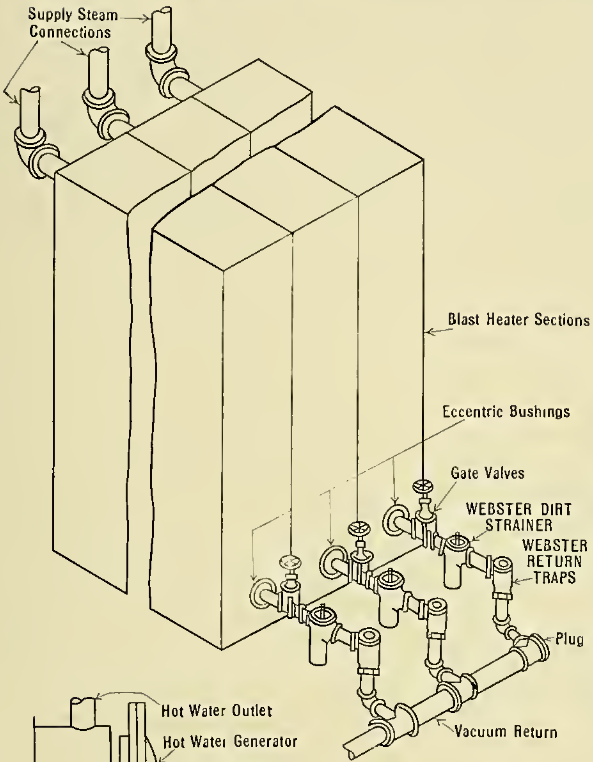


Fig. 22-38. Drip of main and up-feed riser using horizontal cooling surface



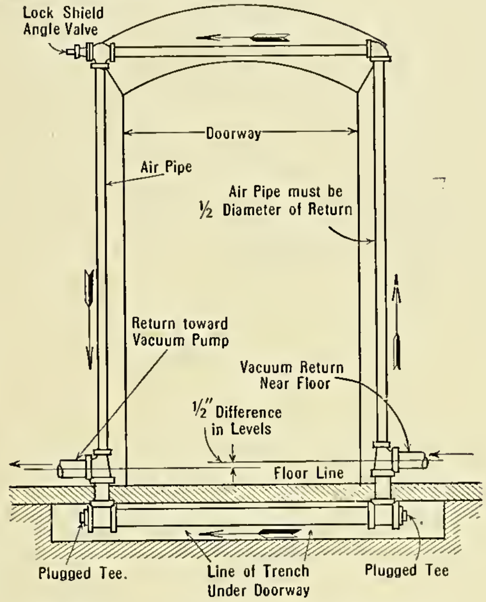


Fig. 22-39. Arrangement of piping where a vacuum return line is carried along the wall near the floor and passes doorways or other openings. The water is carried under the opening and the air is passed through the line over the opening

Fig. 22-40. Method of dripping blast heater section through Webster Return Traps

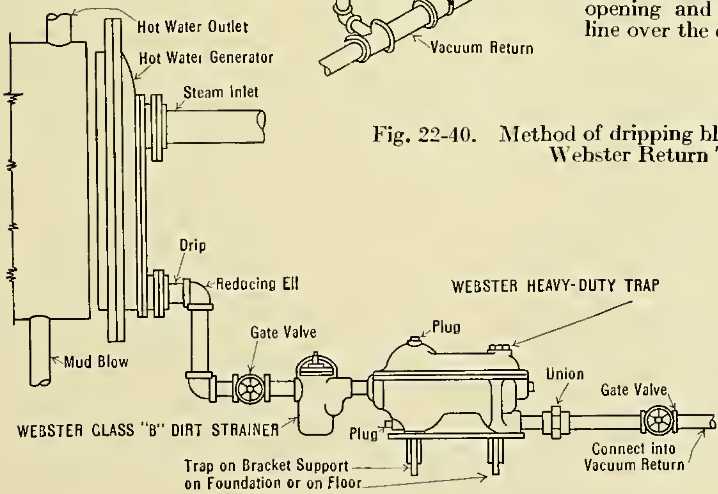


Fig. 22-41. The approved method of draining condensation from the coils of a hot-water service heater to the vacuum return line through gate valve, Webster Dirt Strainer and Webster Heavy-duty Trap

Details Applicable to the Webster Modulation System Only

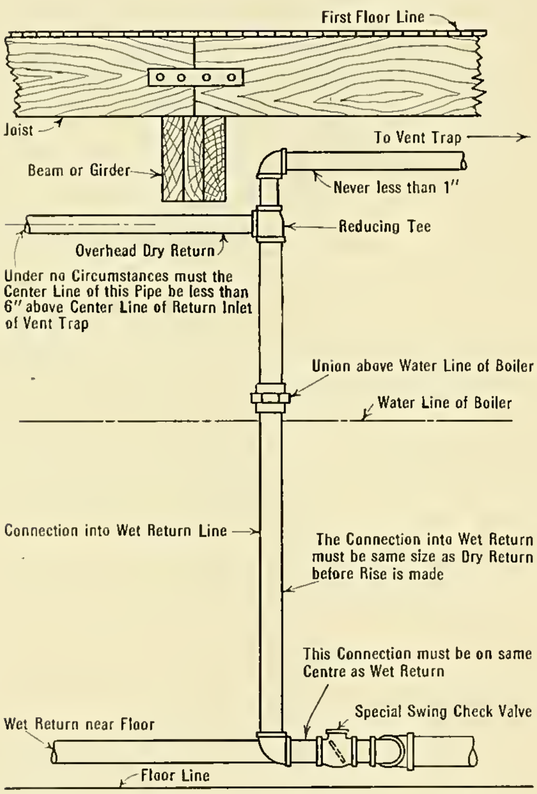


Fig. 22-42. The dry return in a Webster Modulation System, due to its required grade, must sometimes get down into the head room, in which event it may be drained into the wet return and elevated to a higher level. Certain fundamentals must be observed in doing this. The most important is that at the point where the change in elevation occurs, the dry return must never be closer than 6 in. to the level of the inlet to the Webster Modulation Vent Trap

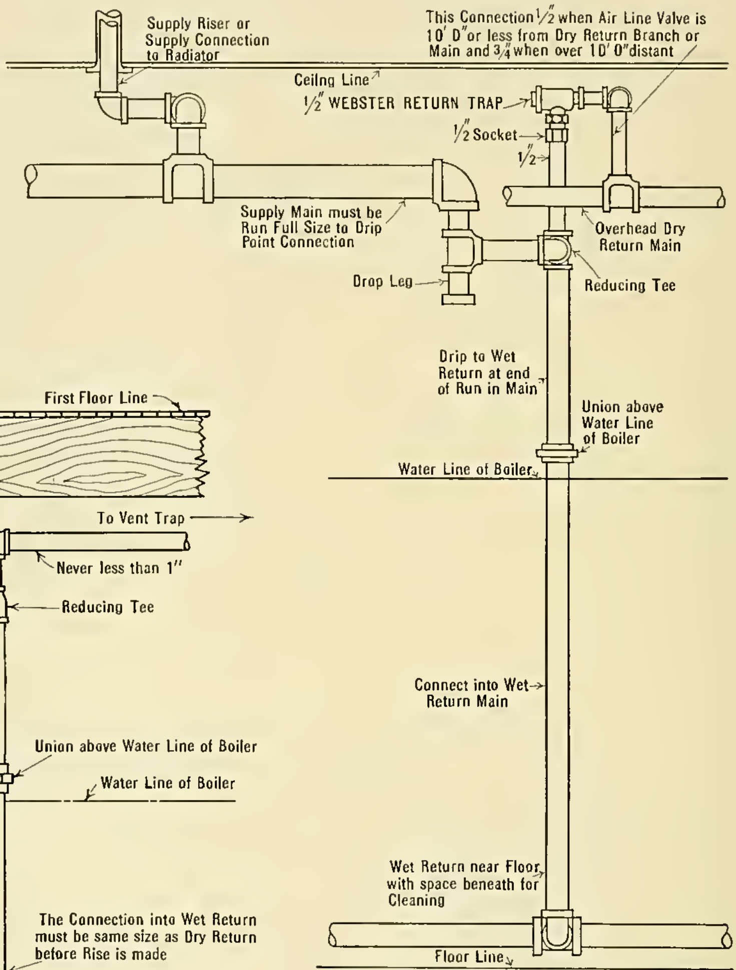


Fig. 22-43. Where a drip is required, at the end of a heating main, the air should usually be vented through a Webster Return Trap into the dry return, as shown in this diagram

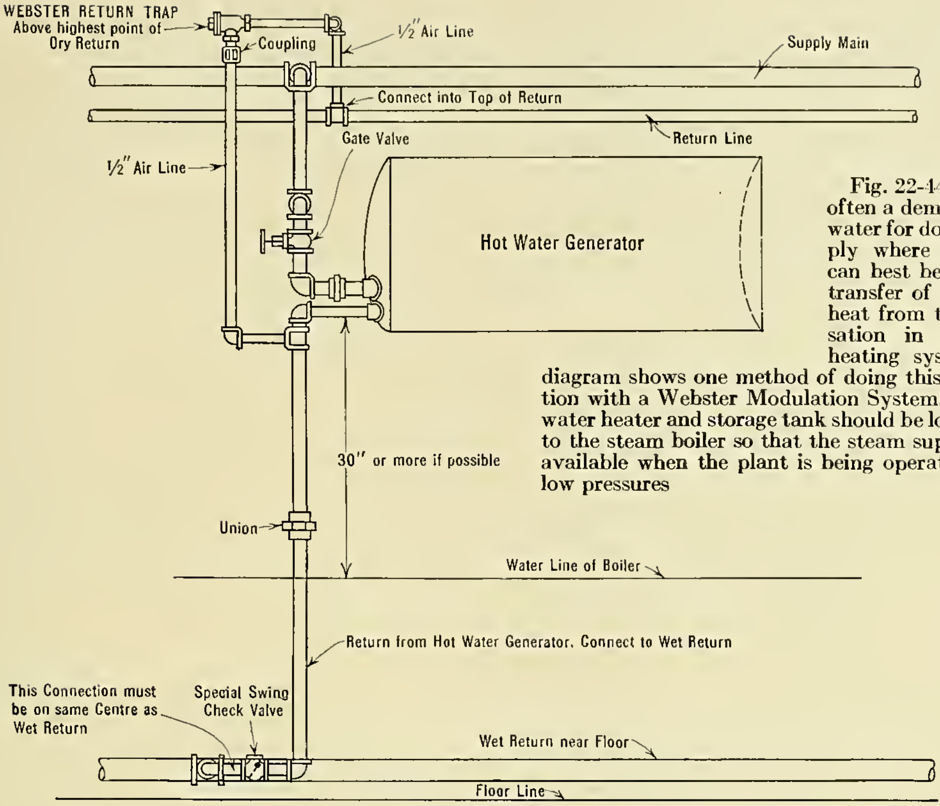


Fig. 22-44. There is often a demand for hot water for domestic supply where this water can best be heated by transfer of part of the heat from the condensation in the steam heating system. The diagram shows one method of doing this in connection with a Webster Modulation System. The hot-water heater and storage tank should be located close to the steam boiler so that the steam supply will be available when the plant is being operated at very low pressures

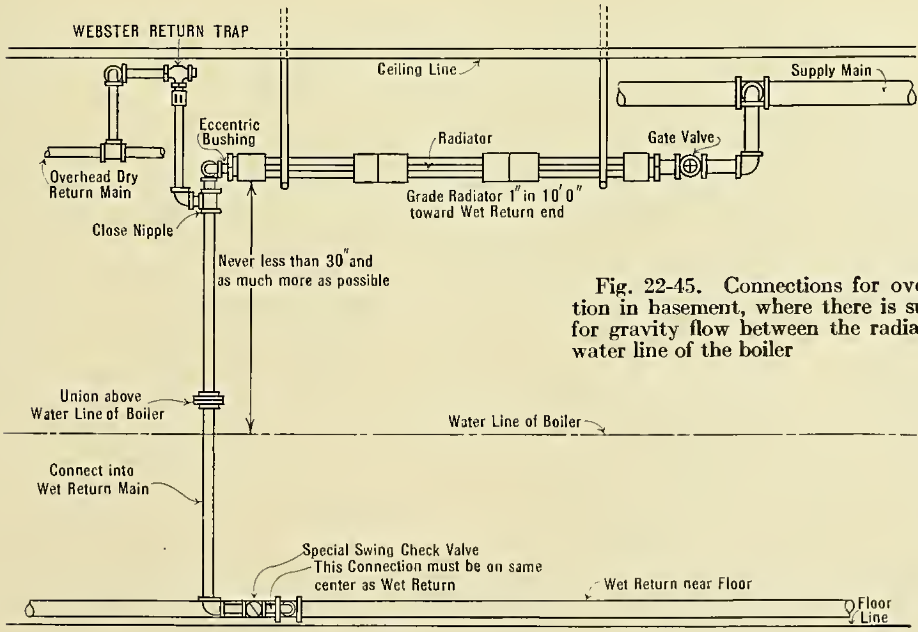


Fig. 22-45. Connections for overhead radiation in basement, where there is sufficient drop for gravity flow between the radiation and the water line of the boiler

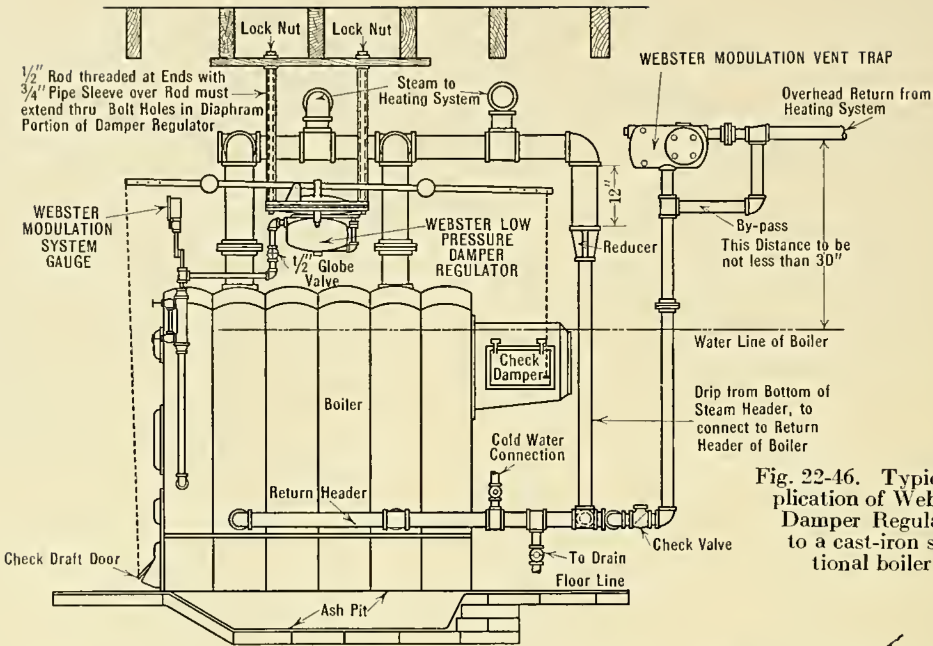


Fig. 22-46. Typical application of Webster Damper Regulator to a cast-iron sectional boiler

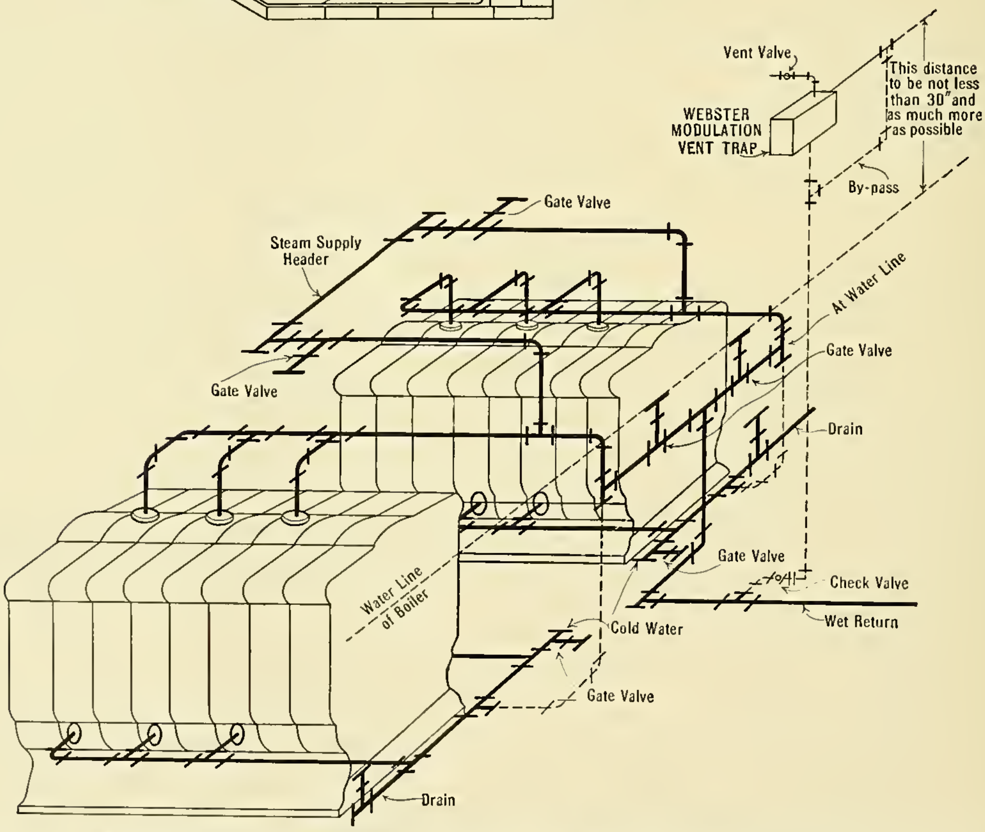
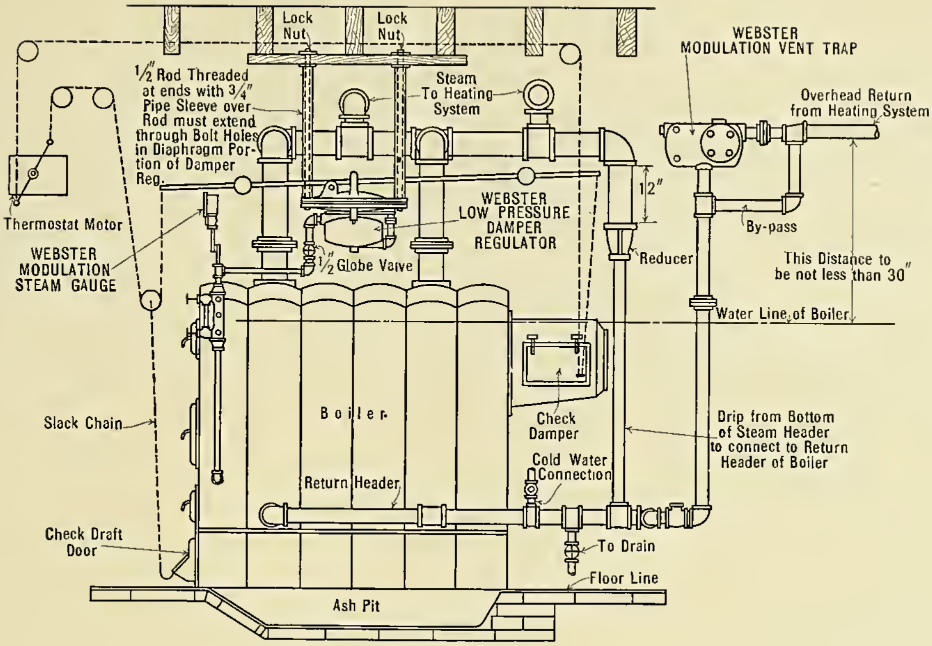
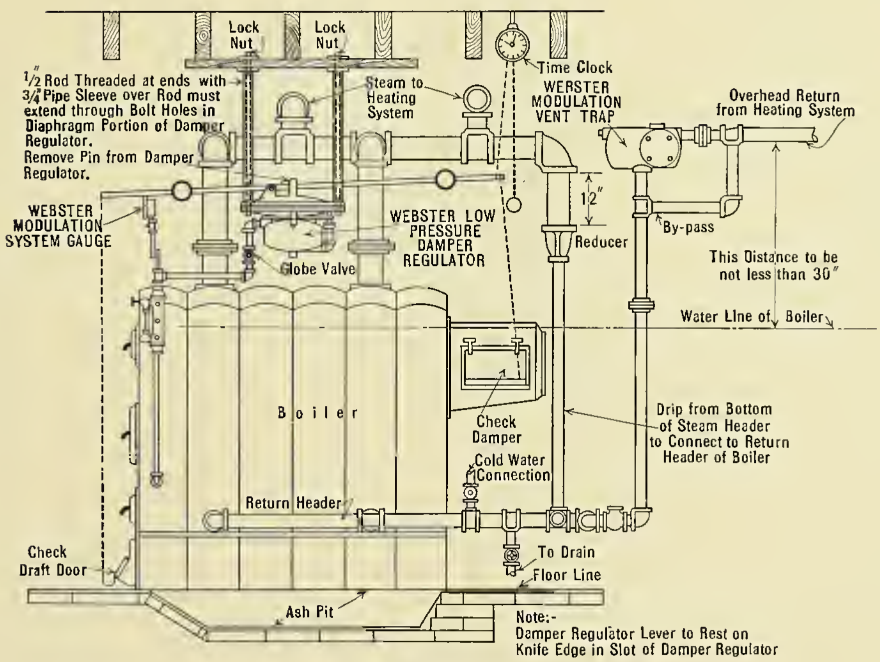


Fig. 22-47. Method of making connections to boilers operating in parallel. Check valve on vent discharge trap only. This is the arrangement of return connections required by many boiler insurance companies



With thermostatic control



With time-clock Control

Fig. 22-48. Typical applications of special controlling devices which may be applied to Webster Damper Regulators

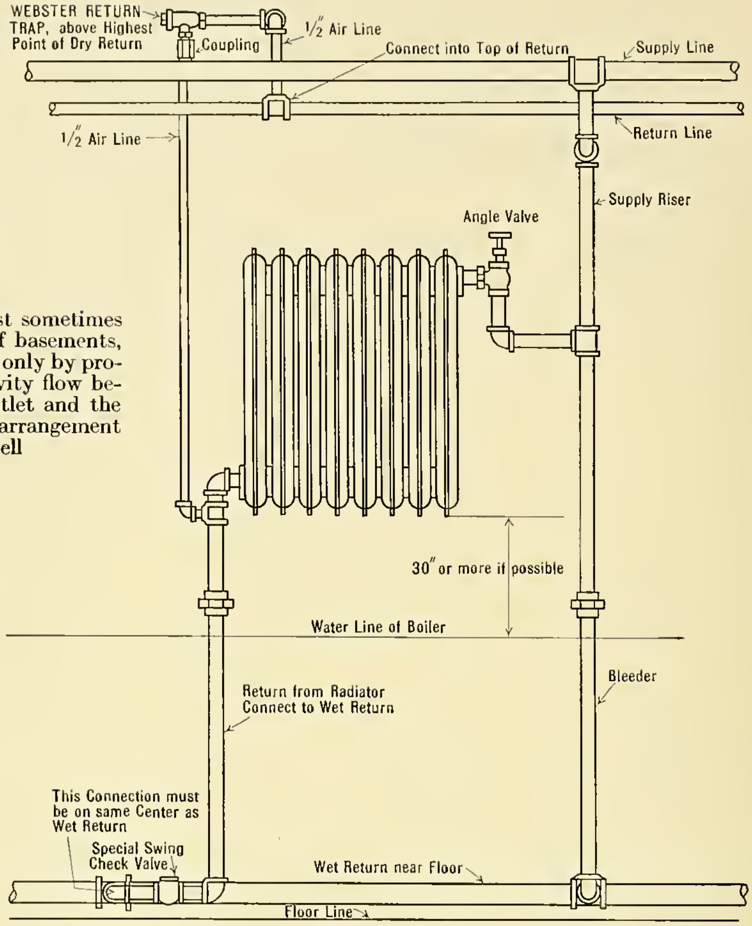


Fig. 22-19. Radiation must sometimes be placed on the side walls of basements, where steam can be circulated only by providing sufficient head for gravity flow between the radiator return outlet and the water line of the boiler. The arrangement shown handles this problem well

CHAPTER XXIII

Capacities and Ratings of Webster Valves and Traps

CAPACITY is a basis obtained from tests under one set of conditions from which ratings are deduced for other operating conditions.

The term *capacity* is used in "Steam Heating" to denote the number of pounds of condensation per hour (W_1) which at uniform flow will pass through the specified apparatus when the pressure is maintained at 1 lb. per sq. in. (P_1) above that of the atmosphere and the pressure at the outlet is that of the atmosphere (P_2).

Having obtained the capacity of any unit of steam-heating apparatus under these standard conditions, *ratings* may be estimated within a very small error, for other stated conditions of pressure difference, time or amount of heat content in the steam at given initial pressure.

For any other pressure difference ($P_3 - P_4$) not differing greatly in amount from the standard pressure difference ($P_1 - P_2$), the quantity of discharge (W_2) varies from the quantity (W_1) discharged under standard conditions in proportion to the square roots of the pressure differences; that is

$$W_2 = W_1\sqrt{\frac{P_3 - P_4}{P_1 - P_2}}$$

or so nearly as to be within the normal errors of test.

The distinction which should be made between *capacity* and *rating*, especially where *rating* is expressed in some indeterminate value like "square feet of radiation," can best be emphasized by examples.

Assume a radiator trap, the capacity of which, with a drop from 1-lb. pressure above atmospheric in the radiator and trap, to atmospheric pressure in the trap outlet, has been found by tests to be 60 lb. of condensation per hr.

Example 1. At what should this trap be *rated* in square feet of radiation on a coil in a room of 60-deg. average temperature, when the steam pressure in the coil is 4-lb. gauge and the vacuum at the trap outlet is 10-in. or 5-lb. gauge?

Answer: The pressure difference through the trap would then be 4 + 5, or 9 lb. The flow through the trap would be as the square root of 1 is to the square root of 9, or three times the capacity of the trap at standard 1-lb. pressure difference. This figures out 180 lb. per hr.

Each pound of steam at 4-lb. gauge pressure gives up in condensing in a coil about 963 B.t.u. of latent heat, a total of 963 × 180 or 173340 B.t.u. per hr. Under the temperature due to 4-lb. gauge pressure the coil would probably give off 324 heat units per sq. ft. of surface. Therefore, the *rating* of this trap under the above conditions would be 324 divided into 173340, or 535 sq. ft. of direct radiation.

Example 2. At what would this same trap be *rated* in *square feet of*

radiation on the same kind of a coil similarly placed when supplied with steam at ¼-lb. gauge, and exhausting to atmospheric pressure at the outlet?

Answer: The pressure difference through trap being as stated, ¼ lb. per sq. in., the flow through trap will be as the square root of 1 is to the square root of ¼, or ½ the rate at 1-lb. difference in pressure, or 30 lb. of steam per hr. Each pound of this steam will give up in condensing about 969 B.t.u. of latent heat or $969 \times 30 = 29070$ B.t.u. per hour.

Under the temperature due to ¼-lb. gauge pressure, the coil would probably give off 300 B.t.u. per sq. ft of surface. Therefore the *rating* of the trap under the conditions of this example would be 29070 divided by 300 = 96.9 sq. ft. of direct radiation.

In Example 1, the *rating* in sq. ft. of radiation is more than five times that in Example 2, the difference being due to the effect of differences in pressure on the same trap, which in both cases had the same *capacity*.

WEBSTER MODULATION SUPPLY VALVES: Careful consideration should be given to the following facts concerning *ratings* of this type of apparatus:

The *capacity* of a modulation valve should be based on the quantity of steam expressed in pounds per hour, or the equivalent B.t.u. of latent heat therein at 1-lb. pressure above atmospheric pressure which will flow through the valve when the outlet is at atmospheric pressure.

This *capacity* may be referred to as the number of square feet of radiating surface which will absorb the total latent heat of the steam flowing into the surface in a given time, at the commencement of which the temperature of the metal of the radiation and the room are at a stated degree below the normal room temperature.

The steam requirements for all types of radiation are greatest during the heating-up period. This is the period during which the cold metal is absorbing heat, while at the same time the radiator as a whole is giving off heat by radiation and convection at approximately one half its normal rate. This statement is approximate because the temperature of the radiating surface is gradually increasing from the cold room temperature to the steam temperature, during this period.

Other things being equal, it follows that the longer the allowable heating-up period, the greater is the proportion of *capacity* which may be expressed in the *rating*. Each type of radiation having a different weight of metal per square foot of heating surface and a different heat emission rate, will take a different *rating* of inlet valve of a given *capacity*.

The consensus of opinion seems to be that the *rating* of a valve should be only such part of its *capacity* as will permit the heating of the entire radiator to steam temperature from a room temperature of 40 deg. fahr. in 20 minutes from the time the valve is fully opened, and this is taken as the heating-up period in the *ratings* given in the tables in this chapter. Radiation, according to type, varies in weight between 2.3 and 7 lb. per square foot of surface.

This causes a marked difference in the steam requirements during the heating-up period, as well as a marked difference in the *rating* of any valve of given *capacity*.

In Table 23-1 the warming-up requirements of the various types of

direct radiation in general use and, in Table 23-2, the normal heat emission in 70-deg. air, have been averaged under five classifications. From these averages, the factors in column 6 have been derived by which the *capacity* of any inlet valve in pounds of steam per hour at 1-lb. differential may be converted into *rating* in square feet of radiation of any of these general classes.

Table 23-1. Basis for Rating Inlet Valves

Heat required to raise temperature of metal from 40 to 210 deg. fahr. in 20 minutes.
Temperature difference 170 deg. fahr. Specific heat, cast iron .12; mild steel .117

Avg. wt. per sq. ft. cast-iron floor radiation	7.00 lb.	x .12	x 170	= 142.80	B.t.u. per sq. ft.
" " " " " cast-iron wall radiation	6.50 "	x .12	x 170	= 132.60	" " " "
" " " " " sheet-steel radiation	2.30 "	x .117	x 170	= 45.75	" " " "
" " " " " 1¼-in. coil radiation	5.20 "	x .117	x 170	= 103.42	" " " "
" " " " " 1 -in. coil radiation	4.85 "	x .117	x 170	= 96.47	" " " "

Table 23-2. Rating Values for Modulation Valves
For various types of direct heating surface

	Col. 1	Col. 2	Col. 3	Col. 4	Col. 5	Col. 6
	B. t. u. per sq. ft. per hr. to maintain 210° in the rad. with room temp. of 70°	B. t. u. emitted in 1-3 hour during warming-up period = 1-6 hourly rate	B. t. u. per sq. ft. to raise temperature of metal	B. t. u. per sq. ft. req'd in 20 minute period. Total of Cols 2 and 3	Combined hourly rate in B. t. u. Col. 4 $\times \frac{60}{20}$	Factor for converting capacity into rating 970 ÷ Col. 5
Cast-iron floor radiation	245	40.8	142.8	183.6	551	1.76 Avg.
Cast-iron wall radiation	296	49.33	132.6	181.93	546	1.78 1.77
Sheet-steel radiation	260	43.33	45.75	89.08	267	3.63
1¼-in. pipe coil radiation	326	54.33	103.42	157.75	473	2.05
1-in. pipe coil radiation	296	49.33	96.47	145.80	437	2.22

To ascertain the *rating* in terms of square feet of radiation of any inlet valve for 20-min. heating-up period, multiply the *capacity* of the valve expressed in pounds of steam per hour at that given pressure difference by the factor in column 6 corresponding to type of radiation and the result will be the square feet of that surface heated from 40 deg. to 210 deg. in 20 minutes.

To ascertain *ratings* for any other period than 20 minutes, a new table must be prepared retaining columns 1 and 3. New column 2 will be determined by multiplying the B.t.u. in column 1 by one-half the selected warming-up period in parts of one hour. (See seventh paragraph, page 234).

New column 4 will be the sum of new column 2 and standard column 3.

New column 5 will be the product of new column 4 by (60 divided by the selected warming-up period in minutes).

New column 6 will be the quotient of new column 5 into the latent heat in 1 lb. of steam at pressure.

Having the *rating* for any particular valve for a particular class of radiation at 1-lb. differential, *ratings* at other pressure differences may be closely approximated by multiplying the 1-lb. rating by the square root of the other pressure difference.

The normal average flow to a heated cast-iron radiator is about 250 B.t.u. A properly designed modulation valve, when 0.6 open should supply the radiator with $\frac{5}{12}$ of the full-open flow, which is the approximate need

for full modulation effect. The balance, or $\frac{7}{12}$ of the opening, is thus available for a quick warming-up period (20 minutes) when the valve is full open.

Owing to the wide difference in area between standard pipe sizes, a valve of say 1-in. size must be used on all different sizes of radiators between its own maximum rating and that of the next smaller, or ¾-in. valve. The wide-open 1-in. valve will therefore produce a much more rapid heating-up effect when connected to a radiator which is a little too large for a ¾-in. valve, and the full modulation effect will be reached much before the valve is 0.6 open, which is the normal position for full modulation effect. This problem might be solved were it not for commercial considerations, by putting a restrictive valve piece in those valve bodies which are used on the lower half of the range. This would limit the flow at 0.6 open to about half way between the maximum for that particular valve and the maximum of the next smaller size. In this way, a valve having a total range of 45 to 78 sq. ft. of radiation at 0.6 open can be limited to 45 to 60 sq. ft. of radiation, thus gaining the whole 0.6 range for controlling the degree of modulating effect, instead of commencing to modulate only after about ⅔ closed and having but the remaining ⅓ of the total movement for graduating the modulating effect.

The *ratings* of each Webster Type W Modulation Valve for the stated conditions, at various positions of the pointer, are indicated in Figure 23-1, which in conjunction with Table 23-3 will assist in selection of a valve of the proper size for any set of conditions.

Initial steam pressure alone is not a correct basis for valve rating or sizing. It is far safer to allow for maximum possible drop in line pressure when figuring the inlet pressure at the valve. Similarly, allowance must be made for variation in return line pressure, especially with vacuum systems.

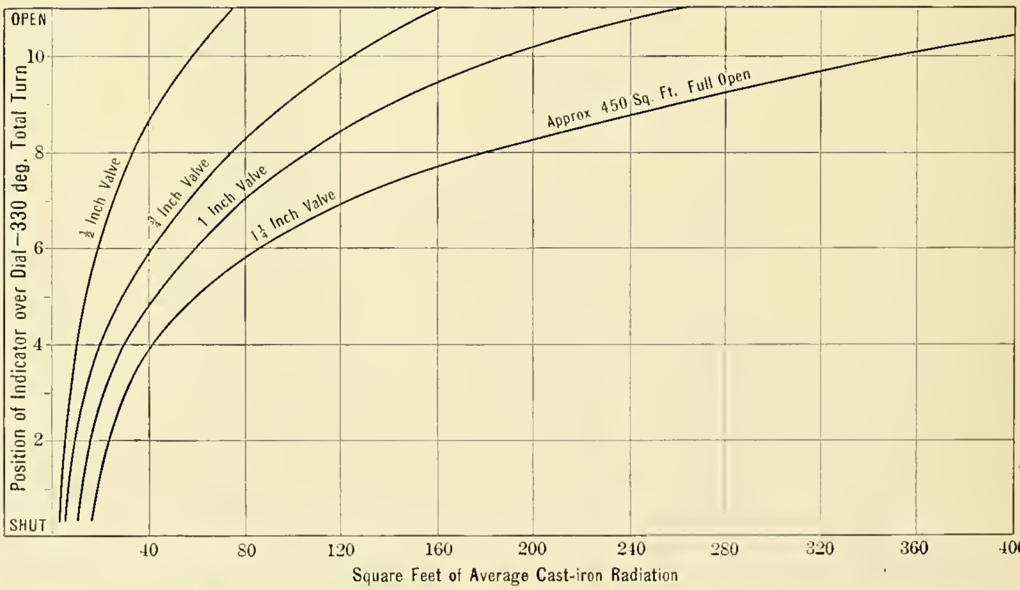


Fig. 23-1. Rating of Webster Type W Modulation Valves. Based upon a differential of *one pound* at the valve and fully heating the radiator in 20 minutes in a room temperature of 40 deg. fahr.

The condensation rate of radiation varies with the type of radiation or coil, its location, and the difference between outside and room temperatures, and allowance must be made accordingly.

Table 23-3. Ratings of Webster Type W Modulation Supply Valves

In square feet of average cast-iron direct radiation at various pressure differences. Based on 20-min. heating-up period from 40 deg. fahr. initial temperature*

Size of valves	Pressure difference					
	1 oz.	2 oz.	4 oz.	6 oz.	8 oz.	1 lb.
	Square feet of average cast-iron direct radiation					
½″	19	27	38	47	54	76
¾″	40	57	80	98	113	160
1″	65	94	132	63	187	265
1¼″	112	160	225	76	319	450

Table 23-4. Ratings of Ordinary Angle-pattern Radiator Supply Valves

In square feet of average cast-iron direct radiation at various pressure differences. Based on 20-min. heating-up period from 40 deg. fahr. initial temperature*

Size of valve	Pressure difference					
	1 oz.	2 oz.	4 oz.	6 oz.	8 oz.	1 lb.
	Square feet of average cast-iron direct radiation					
½″	21	30	42	52	60	84
¾″	44	62	87	107	124	175
1″	77	102	147	180	204	294
1¼″	126	180	252	308	360	504
1½″	187	258	364	446	516	728

Table 23-5. Ratings of Webster Double-service Valves

In square feet of average cast-iron direct radiation at various pressure differences. Based on 20-min. heating-up period from 40 deg. fahr. initial temperature*

Size of valve	Pressure difference					
	1 oz.	2 oz.	4 oz.	6 oz.	8 oz.	1 lb.
	Square feet of average cast-iron direct radiation					
¾″	42	60	85	104	120	166
1″	69	97	138	168	195	275
1¼″	119	168	238	292	336	475
1½″	172	243	343	420	486	685

*If the quick heading-up feature is disregarded and ratings are desired for normal requirements only, after the radiator has been heated up, multiply the values in the tables by 2.2.

Webster Return Traps: Both the Webster Sylphon Return Trap and the Webster No. 7 Return Trap are rated on the basis of the quantity of condensation which they will pass under stated conditions.

Owing to the fact that these traps when cold are fully open, the warming-up period of a radiator has no bearing upon the problem of rating return traps even though the discharge of air and water are then at maximum.

The thermostatically actuated members of Webster Sylphon and No. 7 Return Traps are sensitive to very slight changes of the temperature of

the surrounding medium. The motion of the members is due to the difference in pressure and temperature on a hermetically sealed charge, partially liquid, partially gas and vapor, which responds to changes in temperature with material changes in volume and pressure, and this provides a powerful force to actuate the valve piece.

Table 23-6. Ratings of Webster Return Traps in Pounds of Condensation and B.t.u. per Hour at Various Pressure Differences

Size and type of trap	Pressure difference									
	2 oz.		4 oz.		6 oz.		8 oz.		1 lb.	
	Lb.	B. t. u.	Lb.	B. t. u.	Lb.	B. t. u.	Lb.	B. t. u.	Lb.	B. t. u.
½″-512 & 712	14	13580	19	18430	23	22310	27	26190	38	36860
½″-522 & 722	22	21340	31	30070	38	36860	44	42680	62	60140
¾″-533 & 733	66	64020	94	91180	115	111550	132	128040	187	182390
1″-544 & 744	133	129010	188	182360	230	223100	265	257050	375	363750
1¼″-545 & 745	265	257050	375	363750	459	445230	530	514100	750	727500

Table 23-7. Initial Steam Pressures and Pressure Drops through Supply Pipes, Modulation Valves and Return Traps of the Heating Systems of Different Types of Buildings

Case	Approximate steam pressure in zero weather	Pressure drop through supply piping	Average pressure differential through valves	
			Modulation supply valve	Return trap
A	½ to ¾ lb.	⅛ lb. with minimum run-400 ft.	2 oz.	2 oz.
B	1 to 1½ lb.	¼ lb. with minimum run-400 ft.	4 oz.	4 to 6 oz.
C	1 to 2 lb.	½ lb. with minimum run-400 ft.	4 oz.	4 to 6 oz.
D	1½ to 2 lb.	½ to 1 lb.	4 oz.	4 to 6 oz.
E	1½ to 2 lb.	1 lb.	4 to 6 oz.	8 to 12 oz.

NOTE: In modulation systems in conjunction with low-pressure boilers of limited water capacity, it is essential that the drop in pressure through the system be kept well below the pressure due to the static head between the modulation vent trap and the water line of the boiler. Special apparatus may be provided to return water to boiler where, owing to structural conditions, the above outlined conditions cannot be obtained

Note: Webster Water-seal Traps in the few cases where they are used are rated same as the Sylphon and No. 7 Traps.

Selection of Modulation Supply Valves and Return Traps: For any given installation the choice of the proper sizes of modulation valves and return traps will depend upon the available pressure differential through the valves.

This, in turn, is dependent upon the steam pressure maintained at the boiler and the drop in pressure through the piping system. While it is not possible to lay down hard and fast rules which are applicable for every installation, the following cases are given as representative types of systems in general use. Cases A to D inclusive, given in table 23-7, relate to

modulation systems, with open returns terminating at the boiler in a modulation vent trap or some similar forms of apparatus. Case E is the usual type of vacuum system. The proper sizing of supply and return pipes is explained in detail in Chapter 11 and the pressure drops referred to below are found in Table 11-8.

Case A: Residences and small apartments where the firing is intermittent, frequently extending over eight or perhaps ten-hour periods and where it is necessary to operate at low steam pressure. In mild weather it may be possible to circulate steam through the entire system at or perhaps slightly below atmospheric pressure. In zero weather a pressure will be maintained at the boiler of from ¼ lb. to ¾ lb. depending upon the kind of fuel, length of firing period and condition of fire.

Case B: Very large residences, apartment houses, small offices and public buildings where large size cast-iron sectional or steel boilers are installed, operating at low steam pressure and under the care of a regular attendant, with continuous firing instead of intermittent.

Case C: Schools and similar buildings containing large amounts of indirect radiation where there are periods of interruption in maintaining pressure on the system and where quick circulation is desired when starting.

Case D: Buildings where the pressure is maintained constant by means of a reducing valve and steam is taken at higher pressure either from its own boiler plant or from a street system.

Case E: Office buildings, industrial plants, etc. in which a vacuum system is installed using live steam at reduced pressure, or exhaust steam from engines, pumps and auxiliary apparatus, supplemented by live steam passed through a reducing valve. The steam pressure at the entrance to the supply piping in zero weather will range from 1½ to 2 lb. and the vacuum on the far end of the return line will be approximately 2-in.

Webster Heavy-duty Return Traps: This trap is for use where large quantities of condensation are to be handled at any temperature. It has a cone-shaped float-operated valve piece seating on a sharp-edged orifice, the seat being below the low-water line of the trap. The air entering the trap is allowed to pass to the return line, through a connection controlled by a thermostatically operated trap discharging through a cored passage to the return line. In special cases the opening through the air orifice may be adjusted by hand.

Table 23-8. Ratings of Webster Heavy-duty Traps in Pounds per Hour at Various Pressure Differences Through the Valve

No allowance made for pressure drop in the connecting piping between radiation and trap or from trap through run-out to return

Size of trap	Pressure difference							
	½ Lb.	1 Lb.	2 Lb.	3 Lb.	4 Lb.	5 Lb.	10 Lb.	15 Lb.
0019	700	1000	1400	1700	2000	2200	3150	3900
019	1250	1800	2500	3050	3600	4000	5700	7000
119	2100	3000	4200	5100	6000	6700	9500	11700
219	5600	8000	11200	13600	16000	17900	25300	31100

Webster Series 20 Modulation Vent Traps: Capacities of Series 20 Modulation Vent Traps are based upon the assumption of an air flow of

6000 cu. ft. per hour through a vent orifice of 1 sq. in. area from a pressure of 1 lb. above atmosphere to atmospheric pressure. This quantity is obtained as follows:

Velocity of flow in feet per second is $V = C\sqrt{2\,gh}$, and the quantity in cubic feet per *hour* is $Q = 3600 \times av.$ in which Q is the quantity in cubic feet, c is a constant (0.7), h is the height of a column of air in feet, required to produce a pressure of 1 lb. per sq. in., a is the area of the orifice in square feet, v is the velocity in feet per second and g is 32.17.

1 Lb. of air contains approximately 13.2 cu. ft. For any other pressure difference not varying greatly in amount from the above standard pressure difference, the quantity of discharge will be substantially proportional to the square roots of the pressure difference. Assuming that 50 sq. ft. of cast-iron radiation, with connecting supply pipes, will contain 1 cu. ft. of space, from which the air must be discharged before steam will enter, the following basic data applies for Modulation Vent Traps.

Table 23-9. Basic Data for Modulation Vent Traps

Size of trap	0020	020	120	220	320
Cubic feet of air discharged per hour at 1 lb. differential	85	660	1176	2652	4710
Cubic feet of air discharged per hour at 1 oz. differential	21	165	294	663	1178
Square feet of direct radiation per hour at 1 oz. differential	1050	8250	14700	33150	58900

Referring to page 117, it is to be noted that air vent traps are rated on the basis of flow of initial air from a system in 40 min. with 1-oz. differential pressure through the system. The table below gives the ratings on this basis for which the Webster Modulation Vent Traps should be applied.

Table 23-10. Ratings of Series 20 Modulation Vent Traps

Size of trap	0020	020	120	220	320
Square feet of direct radiation in 40 min. at 1 oz. pressure	700	5500	9800	22100	39265
No. of ½-in. unit vent valves required	1	1	2	3	5

Modulation Vent Valves are required wherever it is desired at times to operate the heating system at a pressure less than atmospheric. Where large heating units are under automatic temperature control, the use of these vent valves is inadvisable unless vacuum breakers are provided at the proper points in the piping system.

CHAPTER XXIV

Appliances for Webster Systems of Steam Heating

WEBSTER Appliances used as parts of heating systems are illustrated and briefly described in the following pages.

These appliances include:

Return Traps
Heavy-duty Traps
High-differential Heavy-duty Traps
Modulation Supply Valves
Double-service Valves
Oil Separators
Grease and Oil Traps
Suction Strainers
Dirt Strainers
Vacuum-pump Governors
Lift Fittings
Return Tanks
Water Accumulators
Vapor Economizers
Gauges
Modulation Vent Traps
Modulation Vent Valves
Damper Regulators
Hylo Vacuum Controllers
Hylo Traps
Conserving Valves
Boiler Feeders
High-pressure Traps
Hydro-pneumatic Tanks
Expansion Joints
Steam Separators
Feed-water Heaters

Return Traps for Automatically Removing Water of Condensation and Air from Heating Units

The return trap, to be perfect in operation, should—

(*a*) Allow the condensation to escape at a temperature slightly below that of the steam.

(*b*) Drain the radiator thoroughly by gravity, without the assistance of pressure or vacuum. A water-logged radiator loses efficiency because part of the heating is being done by the water condensed from steam, which is at lower temperature, and because a water-logged radiator is also an air-bound radiator.

(*c*) Permit continuous removal of air. An air-bound radiator loses efficiency because the steam cannot completely fill it.

(*d*) Automatically close to prevent loss or waste of steam.

(*e*) Work within the widest necessary range of pressure and vacuum variation.

(*f*) Require no adjustment under such variations.

(*g*) Be noiseless in operation, if used where noise is objectionable.

(*h*) Be so designed that the valve will close even where dirt may be present in normal quantities.

(*i*) Be durable and require little or no attention or repairs.

The efficiency of the radiator will depend upon how nearly the return trap meets these requirements.

A return trap working sluggishly will not only hold back the water, but will "bottle up" the air and air-bind the radiator, thus defeating the very purpose of a vacuum system.

As different methods must at times be employed in connection with direct radiators, blast sections, riser drips, main drips, dripping hot-water generators, factory coils, etc., Webster Return Traps are made in several forms, at least one of which will meet the requirements of any installation.

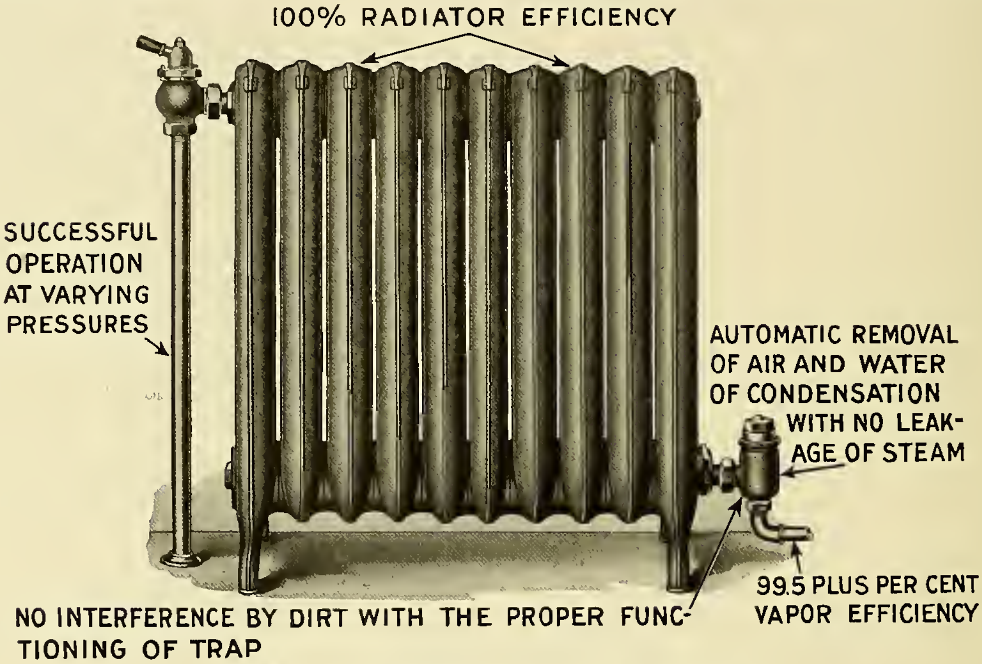


Fig. 24-1. The requirements of a perfect radiator trap

The type and capacity of the trap required depend upon the point of application, the amount of air and water to be removed, the character of the heating surface and the pressure and vacuum carried. It is important that all of these conditions shall be studied carefully before selection is made of the size and type of trap for specific applications.

The Webster Sylphon Trap

The Webster Sylphon Trap has been specially designed to meet the requirements for a perfect radiator trap. It maintains the highest possible efficiency within the heating surface by the removal of all of the products of condensation, and as this is effected without loss of steam, it is economical in the highest degree. The economy is especially apparent when reduced-pressure live steam is used in whole or in part, or where, before its application it has been necessary to waste large quantities of cold water to cool the heating system returns before they enter the vacuum pump.

The operating member consists of a Sylphon bellows, which carries a

conical-shaped valve piece, closing against a sharp-edged seat. The bellows member is very sensitive, operating to close or open the valve port by the slightest change in the temperature of the surrounding medium, and is the most durable form of thermostatic device so far known. The multiple construction of the seamless brass folds forming the bellows distributes the

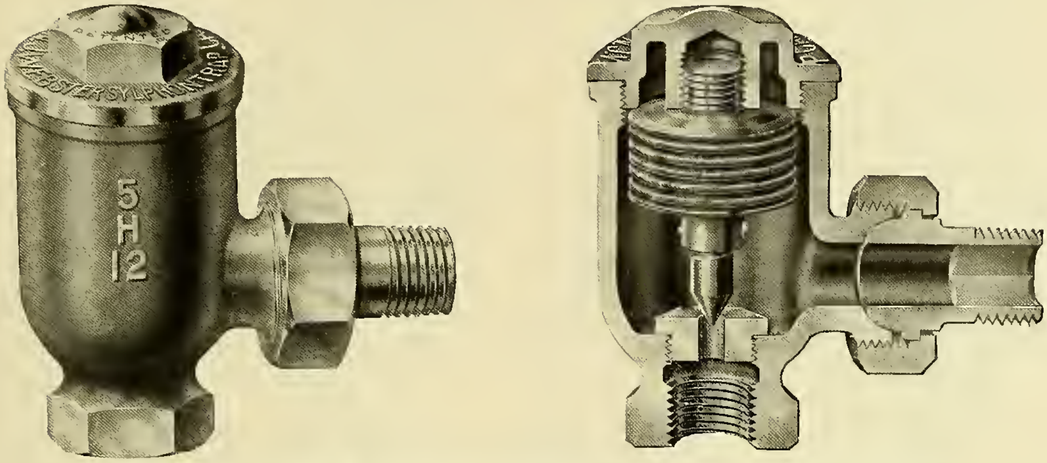


Fig. 24-2. No. 512 Model H Webster Sylphon Trap. Size of pipe connections, ½-in.

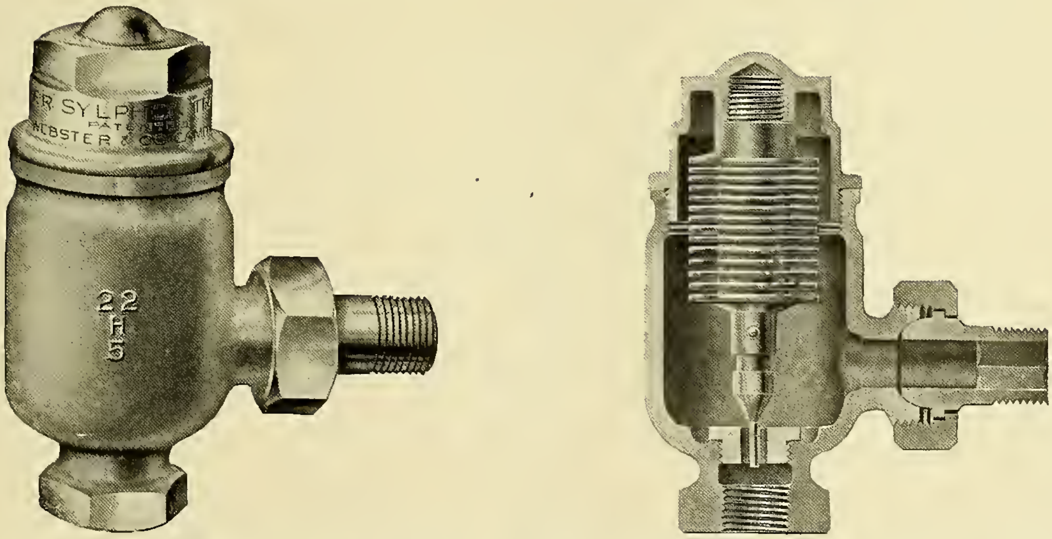


Fig. 24-3. No. 522 Model H Webster Sylphon Trap. Size of pipe connections, ½-in. Nos. 512 and 522 differ in rating and lift of valve, No. 522 being larger
No. 523 has same size body mechanism and rating as No. 522, but has ¾-in. pipe connections to meet unusual specifications in that respect

strain of movement and increases the life of the operating member. Increase in steam pressure on the outside of the bellows is compensated by the increase in pressure on the inside of the bellows.

The sensitiveness of this member is due to the flexibility of the walls

of the bellows to movement in the desired direction and the small amount of movement of each fold when acted upon by the pressure surrounding and also that generated within the bellows. The sum of the small movement of each of the many folds gives a greater total lift of the valve than any other device for similar purpose.

The conical valve piece and sharp-edged seat give increased capacity

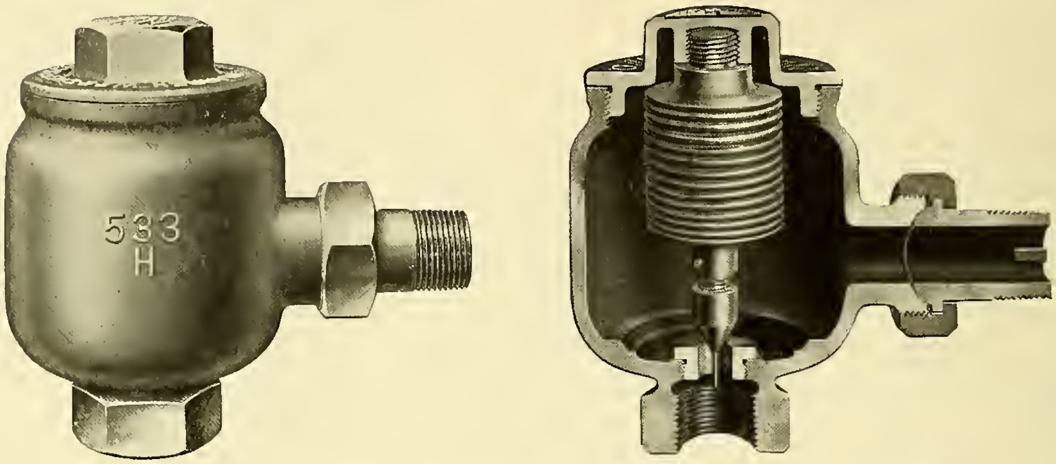


Fig. 244. No. 533 Model H Webster Sylphon Trap. Size of pipe connections, ¾-in.
No. 534 has same size body with 1-in. pipe connections to meet unusual specifications
No. 544 is similar, but larger throughout for 1-in. pipe connections and greater duty
No. 545 is the largest in proportions and ratings. For 1¼-in. pipe connections

for discharge of water, and the valve does not become inoperative due to presence of dirt and scale.

The Webster Sylphon Trap will close quickly and positively when steam reaches the bellows, while the water and air will be freely withdrawn or discharged at temperature slightly below that of steam at existing pressure.

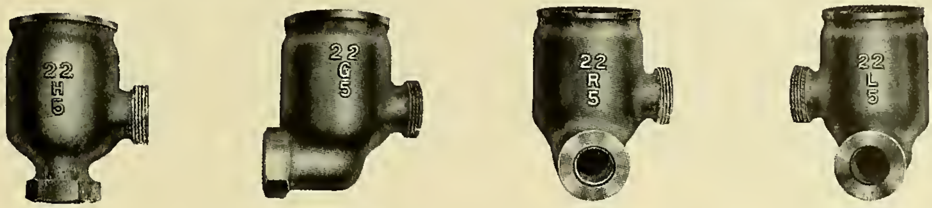
This means that every radiator in use will be thoroughly efficient in heating, as there will be no "pocketing" of air or "bottling up" of water within the radiator.

As the valve is full open when cold, the radiator will be fully drained when steam is turned off, and the vacuum condition existing in the return line will extend within the radiator, assisting circulation when steam is again turned on.

Operation: As the steam first flows into the cool radiator, it expels the contained air and initial condensation through the wide-open trap. As the radiator warms up from inflow of steam, the bellows commences to expand, but remains partially open as long as the air and water in the trap are at a lower temperature than that of the steam. The moment the air is entirely expelled from trap body, and replaced with steam, the valve closes. It opens again when water and air at a temperature slightly less than that of the steam accumulate in the trap. Then, as the water and air escape and are replaced in the trap body by steam, the trap again closes, thus completing its cycle.

Table 24-1. Models and Dimensions of No. 5 Sylphon Traps for Working Pressure Up to 10 Lb. per Sq. In.

For convenience in making pipe connections, Webster Series 5 Sylphon Traps of the smaller sizes are made with four types of bodies as shown. Model H or angle is the one most used



Model H Angle — Model G Straightway offset — Model R Right corner — Model L Left corner

Fig. 24-5. Bodies of Webster Series 5 Sylphon Traps

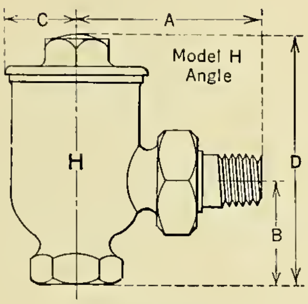


Fig 24-6

Size	Trap no. & model	A	B	C	D
½″	512H	3″	1⅝″	1⅛″	4⅛″
½″	522H	3⅜″	1⅞″	1 3/16″	5¼″
¾″	523H	3⅜″	2″	1 3/16″	5¼″
¾″	533H	4 1/16″	2⅜″	1¾″	5⅜″
1″	534H	4⅛″	2⅝″	1¾″	5 13/16″
1″	544H	4 5/16″	2 9/16″	2″	6 13/16″
1¼″	545H	4½″	2 9/16″	2″	6 13/16″

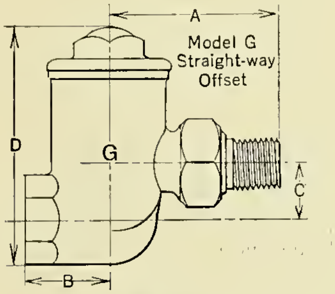


Fig. 24-7

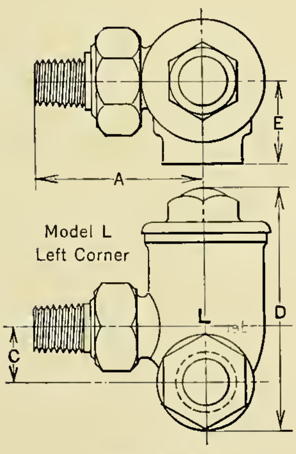


Fig. 24-8

Size	Trap no. and model	A	B	C	D	E
½″	512G, 512R or 512L	3″	1½″	1″	4¼″	1½″
½″	522G, 522R or 522L	3⅜″	1⅝″	1¼″	5⅜″	1⅝″
¾″	523G, 523R or 523L	3⅜″	1 13/16″	1¼″	5 9/16″	1⅞″
¾″	533G	4 1/16″	2 7/16″	1¾″	5¾″	Not
1″	534G	4⅛″	2 7/16″	1¾″	6″	made

For ratings, see Table 23-6, page 238.

The Webster No. 7 Trap

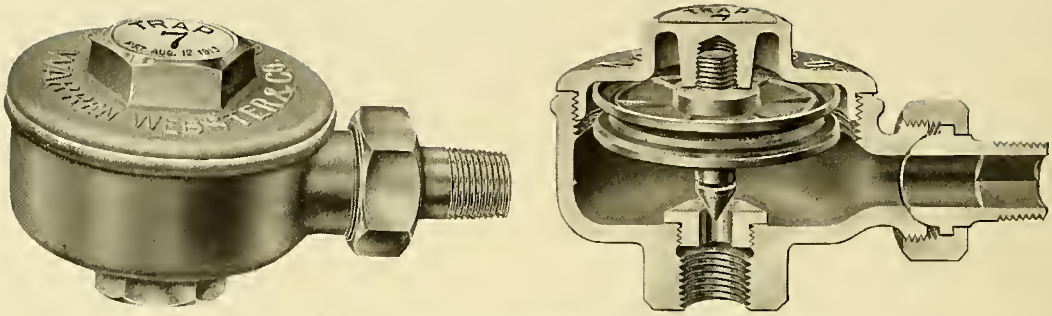


Fig. 24-9 Exterior and interior of No. 722 Webster Trap

Webster No. 7 Traps also realize all of the requirements for thoroughly satisfactory operation as radiator traps. They are applied at the outlets of steam radiators and coils, at drip points on steam supply lines and risers and at the outlets of blast sections on fan coils and provide continuous free and thorough removal of entrained air and water of condensation, without permitting any live steam to escape to waste in the return lines.

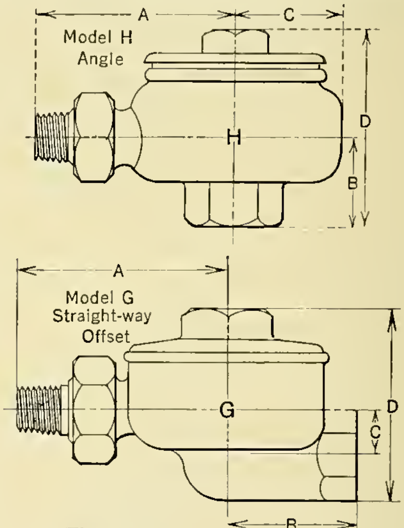


Fig. 24-10

The inlet of the trap is attached to the radiator, coil or supply line by means of the union connection, and the outlet is piped into the return line.

The thermostatic member is inboard of the valve seat where not affected by pressure or temperature in the return line.

The diaphragm, which forms the active part of the operating member, is built of

Table 24-2. Models and Dimensions of Webster Series 7 Traps for Working Pressure Up to 10 Lb. per Sq. In.

For convenience in making pipe connections, Webster Series 7 Traps are made with four types of bodies as shown. Model H or angle is the one most used

Size	Trap no.	A	B	C	D	E
½″	712H	3¼″	$1\frac{7}{16}$″	$1\frac{19}{32}$″	$2\frac{15}{16}$″	
½″	722H	3½″	$1\frac{7}{16}$″	1⅞″	$3\frac{3}{16}$″	
¾″	723H	3⅞″	$1\frac{9}{16}$″	1⅞″	$3\frac{5}{16}$″	
¾″	733H	4¾″	1⅞″	2¼″	$4\frac{9}{16}$″	
1″	744H	4¾″	2″	2½″	$4\frac{9}{16}$″	
1¼″	745H	4¾″	2″	2½″	$4\frac{9}{16}$″	
½″	712G, 712R, 712L	3¼″	2⅛″	¾″	$3\frac{1}{16}$″	2⅛″
½″	722G, 722R, 722L	3½″	2¼″	¾″	3½″	2¼″

For ratings see Table 23-6, page 238

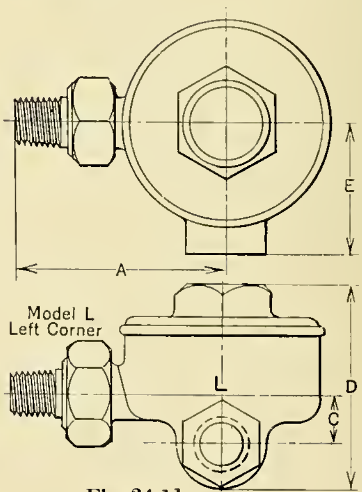


Fig. 24-11

four successive phosphor-bronze plates instead of the usual two and for that reason there is greater diaphragm movement and the valve has greater lift than usually found in traps of similar types.

The expansion and contraction of the diaphragm member is produced by differences in volume and pressure of a hermetically-sealed fluid charge in response to changes in temperature. Even a very slight temperature change produces a powerful force to actuate the conical valve piece, which in closing, fits tightly on a sharp-edged seat.

No part of the valve mechanism is impaired by the quantities of the scale and dirt which normally exist in steam-heating systems.

Webster Heavy-duty Traps

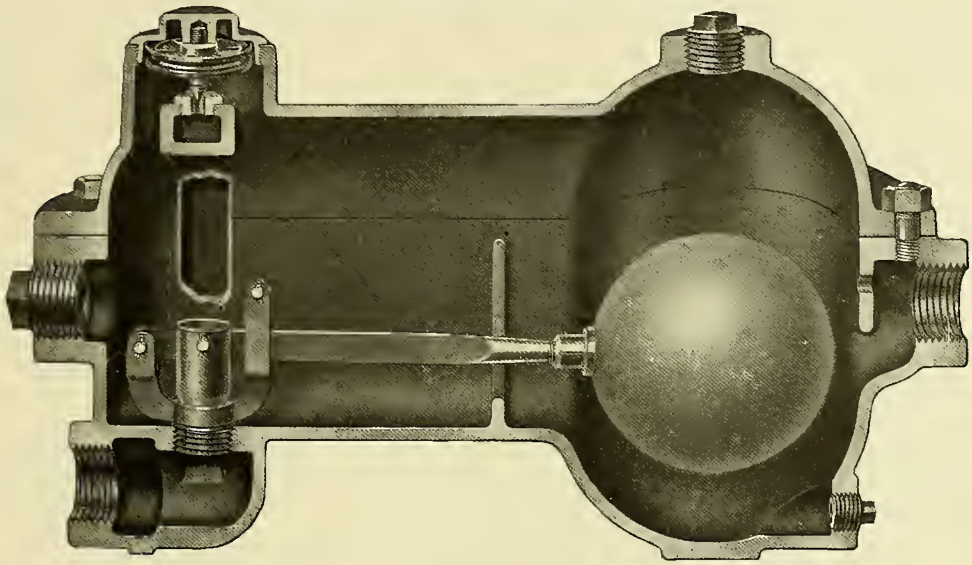
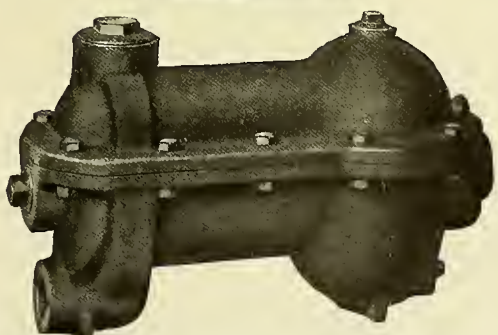


Fig. 24-12. Series 19T Webster Heavy-duty Trap with thermostatically controlled air bypass



Series 19T with Thermostatically controlled Air Bypass. For 15-lb. Maximum Operating Pressure: The Webster Heavy-duty Trap handles unusually large quantities of condensation, and is for dripping main supply risers or mains entering or leaving the building, for draining large sections of blower coils or pipe manifolds, for draining hot-water generators, etc.

Insofar as the discharge of condensation is concerned, this trap operates on the float principle and has a large water outlet to withdraw the condensation as quickly as possible from the unit to be drained.

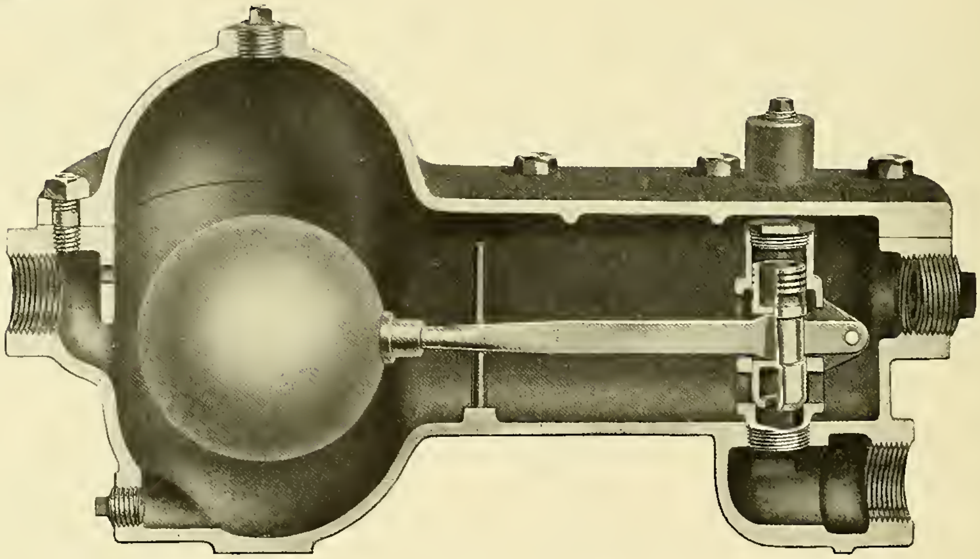
Air is eliminated by means of a thermostatically actuated by-pass, as shown in Figure 24-12. The operating device, the valve piece and seat are the same as used in the Webster No. 7 Trap.

The body and cover are of cast iron. The cover is bolted on, easily removable and so designed that all interior parts are exposed for inspection upon its removal. The outlet is in the bottom of the body, and the inlet may be on either end, with the opposite opening plugged. It is recommended that wherever practical the inlet farthest away from the valve be used. An opening is provided at the bottom of the float chamber as a clean-out by-pass and for draining the trap when out of use.

The float has ample leverage to avoid sticking of the valve. The cone-pointed valve and square-edged seat prevent accumulation of dirt where it might clog the port. The valve is water-sealed at all times, as the water level is always well above the seat. The float lever is kept within the vertical plane of action by guide flanges cast into the trap body.

This trap can also be furnished special with hand-controlled air and by-pass, where unusual conditions require such construction. In such cases the air port is adjustable for any desired degree of constant leakage.

Some of the many practical applications of the Series 19T Trap will be found in Chapter 22. Ratings are given on page 239 and dimensions on page 249.



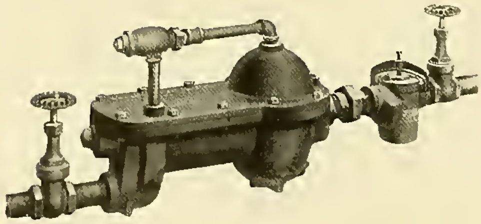


Fig. 24-13. Conventional arrangement of Series 20 Webster High-differential Heavy-duty Trap and Special Webster Dirt Strainer (Inlet pipe may be connected to opposite end if desired)

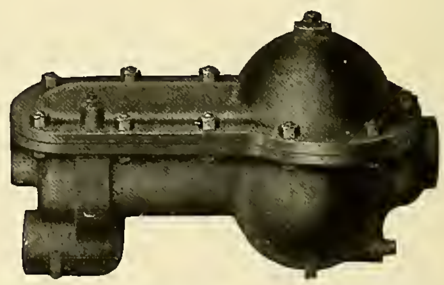


Fig. 24-14. Series 20 Webster High-differential Heavy-duty Trap for working pressures up to 50 lb. per sq. in.

High-Differential Type, Series 20, For Working Pressures up to 50 lb. per sq. in.: The Webster High-differential Heavy-duty Trap is recommended for steam pressures higher than 15 lb. and where large quantities of condensation may be discharged. It is particularly applicable to problems like or similar to those described in Chapter 19.

The trap body is constructed of cast iron and has an easily removable cover of the same material. The valve is of the balanced type and operates against a steam-brass seat. The ball float is extra heavy to withstand the higher pressures.

The Webster High-differential Heavy-duty Trap may be operated with a constant leakage through a hand-adjusted air vent, though the best practice calls for control of the air discharge by means of a thermostatically actuated valve in a by-pass of pipe and fittings as shown in Figure 24-16. It is important to note in this case of higher than ordinary steam pressure, that the thermostatic trap must be of the No. 8 Sylphon type. (See page 275.)

Table 24-3. Dimensions of Webster Heavy-duty Traps

All dimensions in inches and subject to slight variation

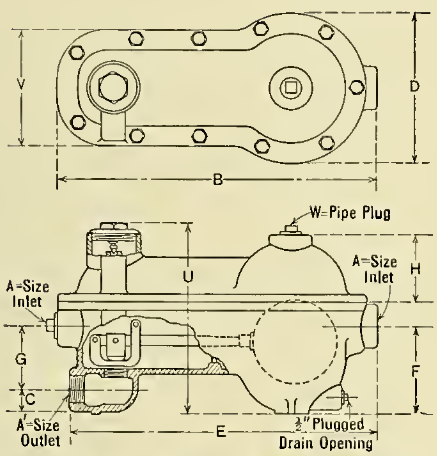


Fig. 24-15. Standard type—Series 19T

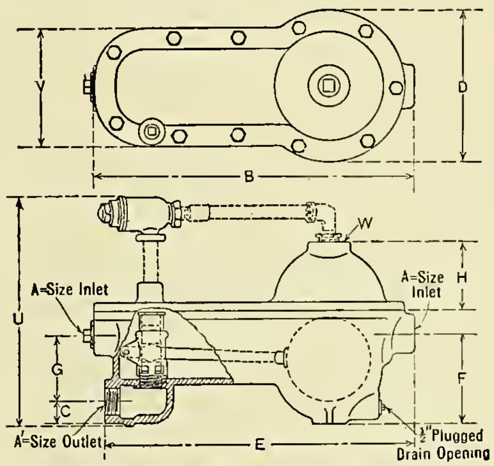


Fig. 24-16. High-differential type—Series 20

For ratings of Heavy-duty Traps see Table 23-8, page 239

Series 19T, with thermostatically controlled by-pass

Number	A	A¹	B	C	D	E	F	G	H	U	V	W
0019-T	¾	¾	13¼	1	7½	12⅝	4⅛	3⅛	2½	9¼	5⅝	½
019-T	¾	¾	15¾	1	8	15	4⅛	3⅛	2⅞	9¾	6¼	¾
119-T	1¼	1¼	19⅛	1⅜	9	18⅜	5⅜	3⅞	4⅛	11½	7	1
219-T	2	2	20⅝	1⅝	10½	19⅞	6⅛	4⅝	4⅜	13¼	8	1½

Series 20, high-differential type

Number	A	A¹	B	C	D	E	F	G	H	U	V	W
020	¾	¾	15¾	1	8	15	4⅛	3⅛	2⅞	12¼	6¼	¾
120	1¼	1¼	19⅛	1⅜	9	18⅜	5⅜	3⅞	4⅛	13¾	7	1
220	2	2	20⅝	1⅝	10½	19⅞	6⅛	4⅝	4⅜	14⅞	8	1½

The Webster Type W Modulation Valve

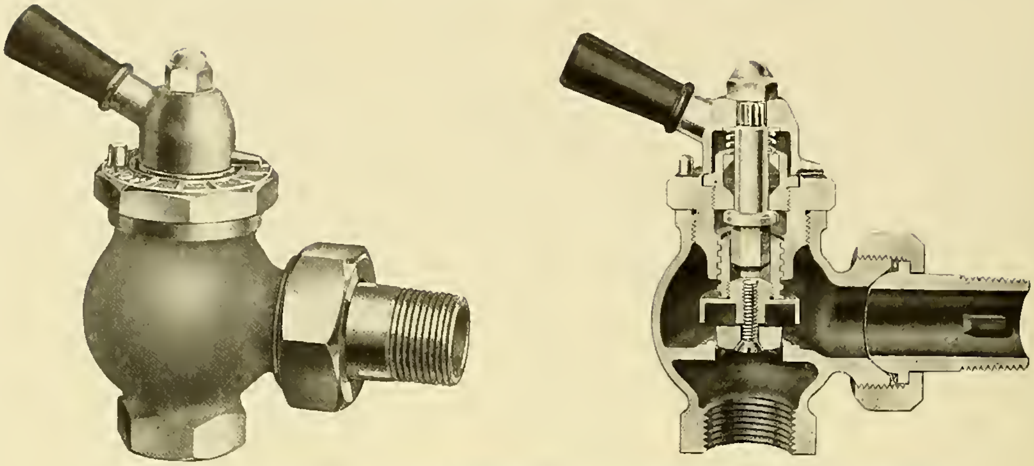


Fig. 24-17. The Webster Type W Modulation Valve—shown in partly open position

The Webster Type W Modulation Valve is a special-purpose radiator valve of the quick-opening, non-rising stem, straight-lift type, built for complete opening or closing with less than a single turn of the handle. Its manipulation is as simple and its control as effective as the movement that regulates light from a gas jet.

As the names implies, the principal function of the Webster Modulation Valve is to facilitate "modulation" of temperature in each room according to the desires of the occupant, by varying the amount of steam admitted to the radiator or coil. A pointer attached to the handle traveling over a graduated dial indicates the amount of valve opening at all times.

With the valve full open, the discharge capacity through the ports is nearly equal to that of the outlet connection of the valve.

Less than three-fourths of the valve lift and opening movement is required to produce modulation up to normal full heating requirement. The rest is in reserve to admit more steam during the heating-up period, as needed to compensate for the higher condensation rate caused by contact with the cold radiator and its surrounding air.

Construction Details: The modulation effect is produced by a patented modulating plug which varies admission of steam in progressive volume with the lift of the valve piece.

A Jenkins disc is used to insure tight closing. With the exception of this and the handle, all parts are of brass. The handle is of special composition and so formed that the hand of the operator does not come into contact with the heated surface of the valve body.

Application: The Webster Modulation Valve may be used on either hot-water type radiators (having connections from section to section at both top and bottom) or with steam type radiators (bottom connections only), although the former type is preferable from the standpoint of convenience.

Where the Webster Modulation Valve is used with the hot-water type of radiator, it should be placed at the top to bring the operating handle in

the most convenient location and to permit the steam to circulate across and downward. Air and condensation, being heavier, fall to the bottom in advance of steam and give full efficiency to the heated part of the radiator.

Where the Webster Modulation Valve is used with a steam type radiator, it is possible by the use of an inlet section of the hot-water type to secure the convenience of operation which is obtained where the valve is placed at the top of the radiator.

If placed at the bottom of radiators, because other connections cannot be arranged, the inlet bushing should be eccentric and so located that the center line of the radiator or inlet is above that of the radiator outlet. This is essential to prevent condensation from draining by gravity through the supply instead of the return connections, thus eliminating water-hammer.

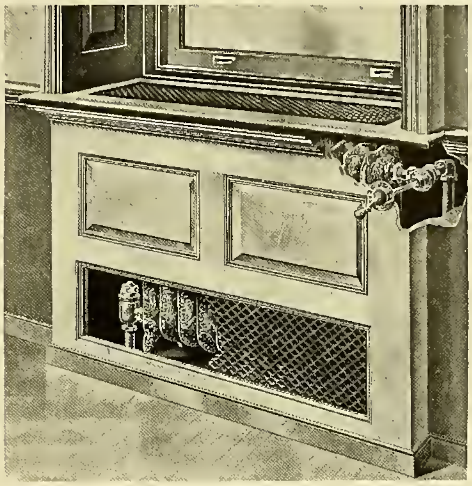


Fig. 24-18. Typical application of the extension stem principle

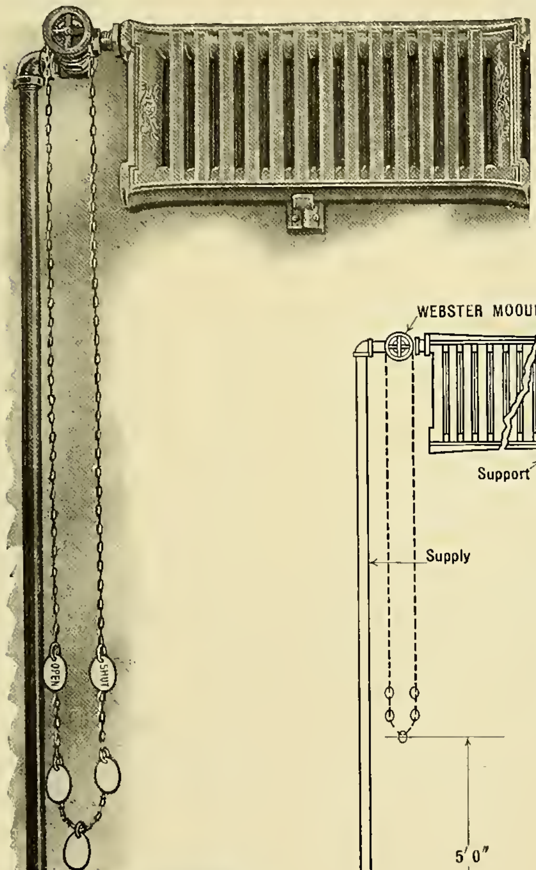


Fig. 24-19. Typical application of chain attachment to Webster Type W Modulation Valve

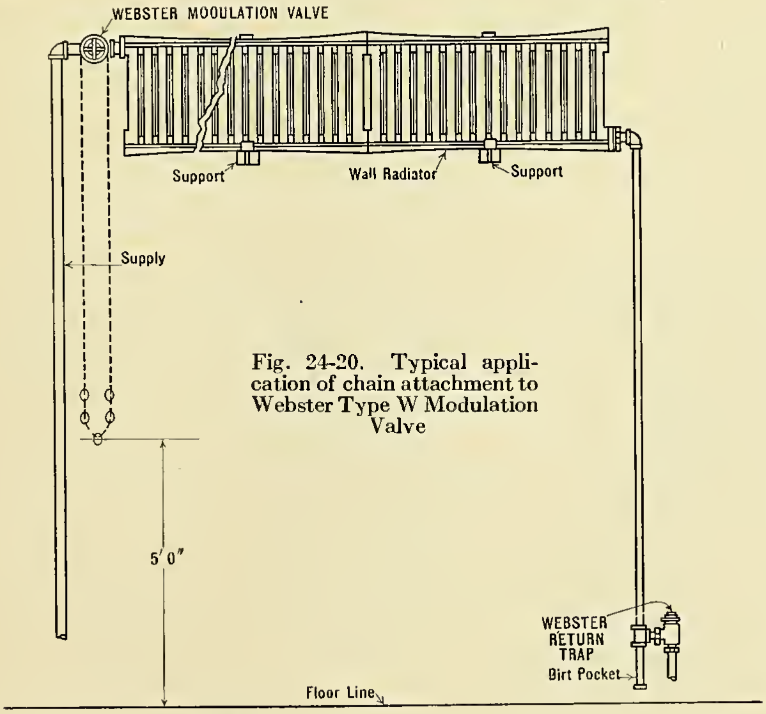


Fig. 24-20. Typical application of chain attachment to Webster Type W Modulation Valve

Extension Stem: For attachment to radiators concealed in recesses or under window seats behind grilles, the Webster Modulation Valve is provided with an extension stem and a special dial that may be placed on the face, top or end of the grille or seat (see Figure 24-18).

The stem has a universal joint on each end, which permits operation of the valve from a point not directly in line with the valve stem, and at the same time provides enough play to avoid sticking or binding from misalignment or shifting caused by expansion and contraction. This construction also avoids the necessity for very accurate stem connections.

The outside indicator dial, pointer and handle are similar to those used on top of the standard valve.

Chain Attachment: The Webster Modulation Valve to be applied to radiators or coils located in skylights, overhead, or on walls near the ceiling, can be fitted with a chain attachment for convenience in obtaining every advantage of the modulation feature (Figures 24-19 and 24-20).

The chain wheel is substituted for the handle of the standard type of Modulation Valve and the chain is made just long enough to permit easy grasp from the floor. Tags are attached to bottom of the chain in such positions that the hanging end indicates the degree of valve opening.

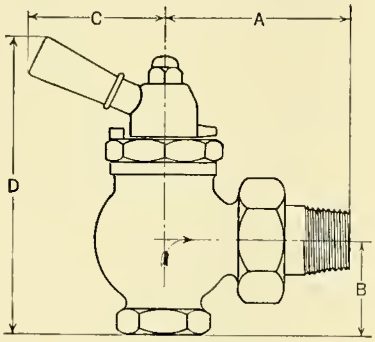


Fig. 24-21

Table 24-4. Dimensions of Type W Modulation Valve

Size	A	B	C	D
1/2	2 3/4	1 1/4	2 1/8	4 1/8
3/4	3 1/8	1 1/2	2 1/8	4 5/8
1	3 1/4	1 3/4	2 1/4	5 3/8
1 1/4	3 5/8	2	2 1/4	6

All dimensions in inches and subject to slight variation. For ratings, see Table 23-3, page 237

The Webster Double-service Valve

This is one of the latest developments of apparatus for simplifying piping connections in steam heating systems in certain types of construction.

Common practice in buildings of only one story and in some other instances calls for a steam supply line along the ceiling of the first floor to feed each radiator or coil through a short down-feed riser, which must be dripped into the return line. This multiplicity of unsightly connections is simplified by the use of Webster Double-service Valves, applied in the manner shown in Figure 24-23.

This valve performs "double service," as a supply valve for the radiator and as a trap for draining the riser.

The thermostatically controlled valve is open when there is water or air in the riser, and permits the condensate to flow through a bypass in valve body into the radiator and thence into the return. Upon presence of steam the thermostatic member expands, closes the valve, and thus prevents waste of steam.

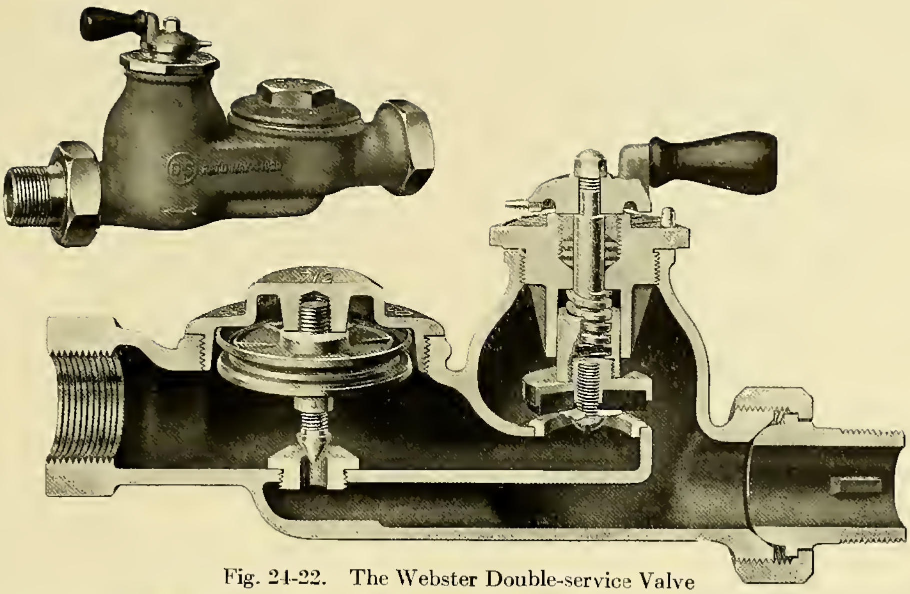
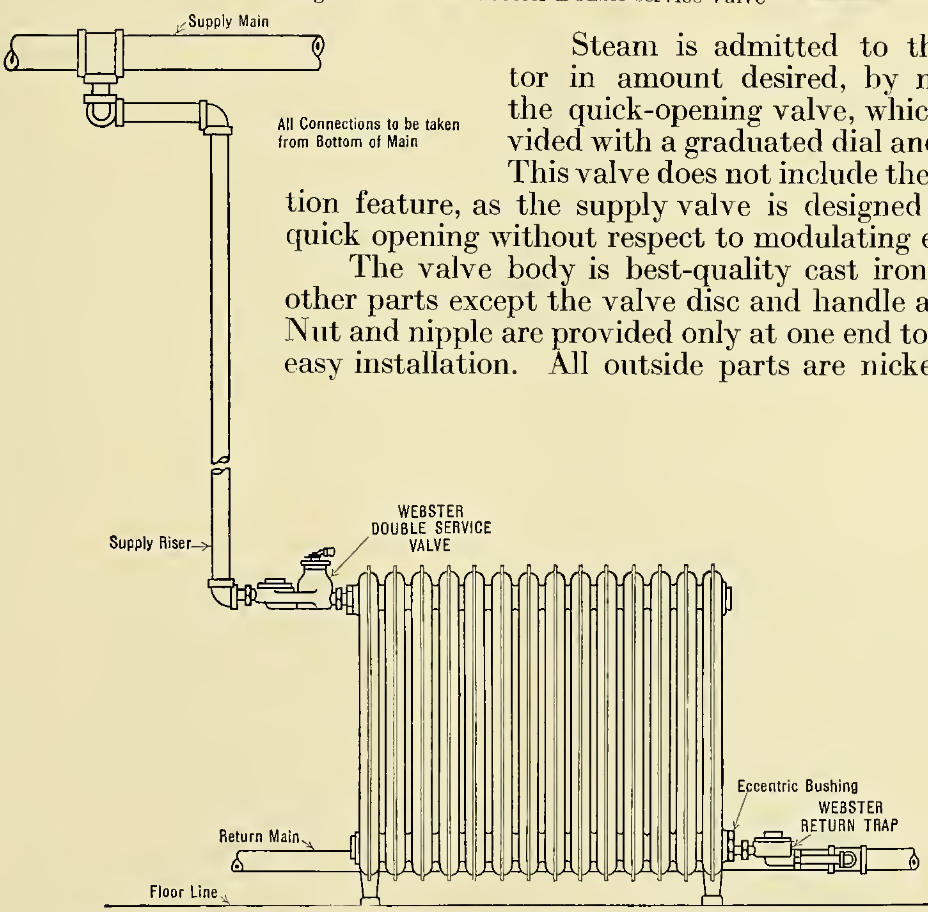


Fig. 24-22. The Webster Double-service Valve



Steam is admitted to the radiator in amount desired, by means of the quick-opening valve, which is provided with a graduated dial and handle. This valve does not include the modulation feature, as the supply valve is designed only for quick opening without respect to modulating effect.

The valve body is best-quality cast iron, and all other parts except the valve disc and handle are brass. Nut and nipple are provided only at one end to promote easy installation. All outside parts are nickel-plated.

Fig. 24-23. Application of a Webster Double-service Valve to a standard cast-iron radiator

The thermostatic member, which is built up of four discs of phosphor bronze and filled with a volatile fluid, the conical valve piece and the sharp-edged seat are of standard pattern as used in the Webster No. 7 Trap.

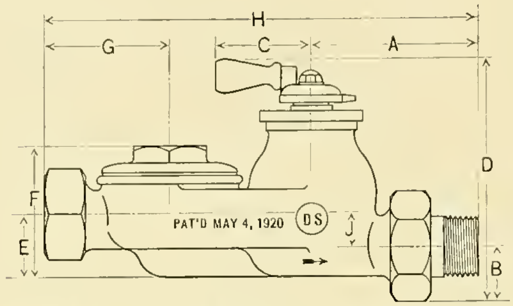


Fig. 24-24. The Webster Double-service Valve

The inlet valve is provided with a ring seat and Jenkins disc to insure tight closing. Its quick-opening feature is provided by a screw stem of such pitch that the valve will be completely opened with less than a complete turn of the handle.

Table 24-5. Dimensions of Webster Double-service Valves

Size	A	B	C	D	E	F	G	H	J
¾	3¼	1	2⅛	5⅜	1⅜	2¾	2⅞	9⅛	⅞
1	3⅝	1¼	2⅛	5½	1½	3	3	9⅝	⅞
1¼	4	1½	2⅞	6¾	1¾	3⅛	3⅛	10½	⅞
1½	4½	1⅝	2⅞	8	1¾	3⅜	3⅛	11¼	⅞

All dimensions in inches and subject to slight variation. For ratings, see page 237

Webster Oil Separators

The Series 21 Webster Oil Separator is made in two patterns—for either horizontal or vertical direction of steam flow. The baffles in the horizontal type are double-hooks so that either nozzle may be used as the steam inlet. The vertical pattern is suitable for up-flow of steam only.

An outstanding feature of this series of Webster Oil Separators is the position of the manhole cover which makes it possible to inspect or clean the device without disturbing the piping.

Separation of oil and condensation is effected by impact upon and adhesion to baffles and by abrupt changes of direction of flow through the separator.

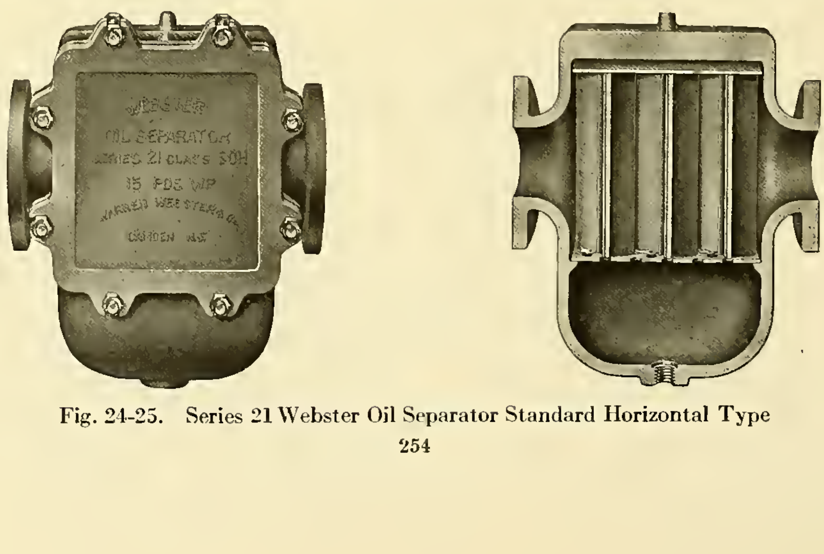


Fig. 24-25. Series 21 Webster Oil Separator Standard Horizontal Type

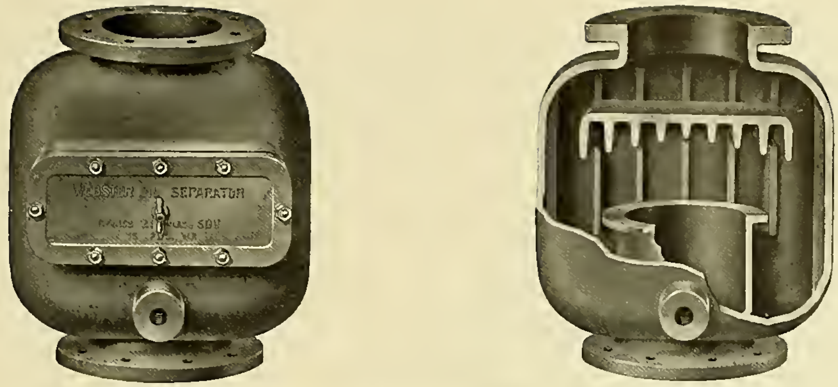


Fig. 24-26. Series 21 Webster Oil Separator, Standard Vertical Type, for upflow only

There is no unobstructed path through any Webster Oil Separator, yet the free area through which steam must pass is several times greater than inlet and outlet area, thus minimizing pressure loss due to friction.

The use of these separators protects boiler heating surfaces and interior surfaces of heating systems from the oil deposits that otherwise seriously impair heat transmission and often cause serious damage.

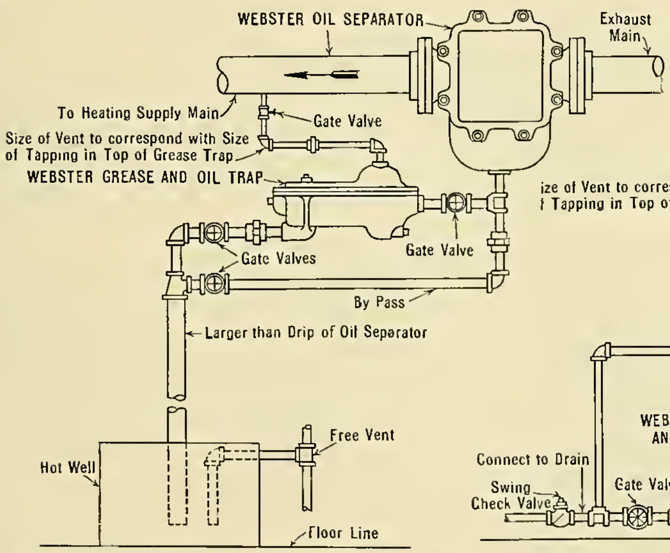


Fig. 24-27. Method of connecting a Webster Grease Trap to a Webster Oil Separator, where a partial vacuum may at times be carried on the heating main

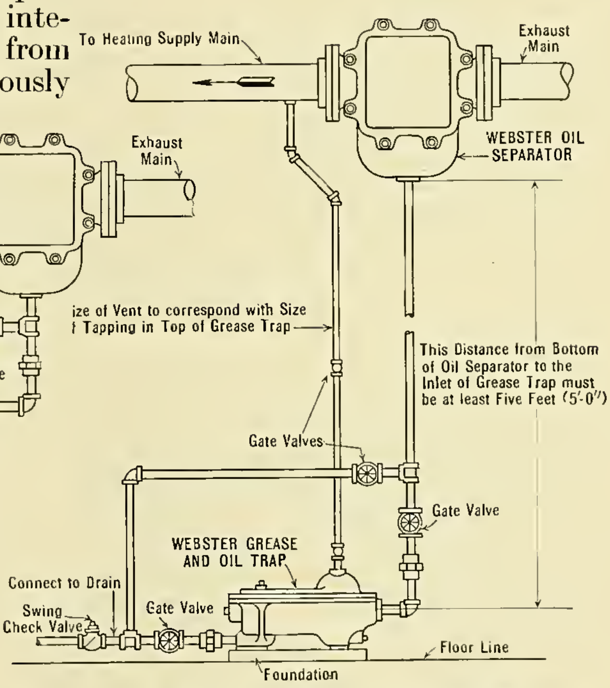


Fig. 24-28. Typical method of draining Webster Oil Separator through a Webster Grease Trap, where positive pressure is maintained at all times

These separators may also be used for such special purposes as removing moisture or oil from compressed air and other gases.

That Webster Separators are efficient in all their standard and special forms is indicated by absolute satisfaction in over 15,000 installations.

The material ordinarily used in the shells is close-grained cast iron, but spec al shell of semi-steel, cast steel or other material can be furnished at extra cost.

Table 24-6. Maximum Ratings of Oil Separators in Lb. per Min. at Average Gauge Pressures Based on 6000 Ft. per Min. Pipe Velocity

Size	Pressure, lb. per sq. in. 0	5	10	15
2	5.2	6.7	8.4	10.
3	11.4	15.	18.6	22.
4	19.8	26.	32.	38.
5	31.	40.6	50.2	59.7
6	45.	59.	73.	86.5
8	78.	102.	126.	150.
10	123.	160.	200.	235.
12	176.	231.	285.	339.
14	222.	292.	361.	427.
16	294.	385.	475.	565.
18	375.	492.	608.	720.
20	452.	595.	735.	870.
22	550.	725.	900.	1060.
24	660.	870.	1070.	1270.

For lower velocities, the pounds carried will be proportional as the lower velocity is to 6000

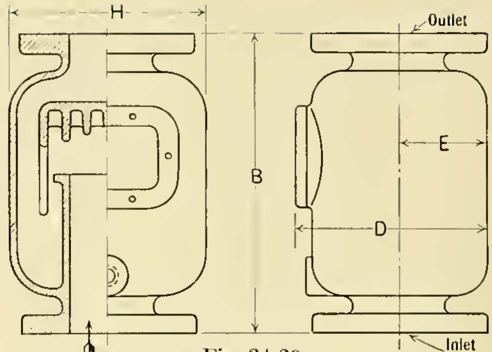


Fig. 24-29

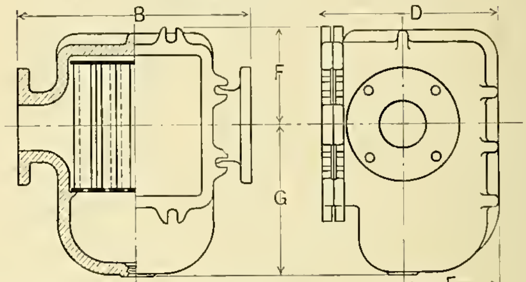


Fig. 24-30

Table 24-7. Dimensions of Webster Oil Separators

All dimensions in inches. Companion flanges furnished only on special order; drilled low-pressure standard unless otherwise ordered

Standard Horizontal Type (Fig. 24-30)—for steam flow in either direction

	Dimensions						Flanges		
SIZE	B	D	E	F	G	Drip	Outside diameter	Bolt circle	No. & sizes of bolts
*1½	10	6⅝	3½	4⅜	6⅛	¾			
*2	10¼	8	4⅜	5⅛	7¼	¾			
2	12	8	4⅜	5⅛	7¼	¾	6	4¾	4–⅝
2½	13¾	10¼	5⅝	6	8⅛	¾	7	5½	4–⅝
3	15	11½	6⅛	6½	9½	¾	7½	6	4–⅝
3½	15¾	10⅜	5	6¾	10½	1	8½	7	4–⅝
4	16½	11¼	5½	6¾	11¼	1	9	7½	8–⅝
5	17¾	11⅝	5¾	7½	13¼	1	10	8½	8–¾
6	19	12⅛	6	8¼	13⅜	1	11	9½	8–¾
8	21	12⅞	6⅝	8⅜	18⅝	1¼	13½	11¾	8–¾
10	22	16	8½	9⅜	19¾	1½	16	14¼	12–⅞
12	24⅛	18¾	9⅞	10½	22½	2	19	17	12–⅞
14	28	22	11½	11¼	22⅞	2	21	18¾	12– 1
16	31	25¾	13½	13	23½	2½	23½	21¼	16– 1

*Screw connections only.

Standard Vertical Type (Fig. 24-29)—for up-flow only

	Dimensions					Flanges		
SIZE	B	D	E	H	Drip	Outside diameter	Bolt circle	No. & sizes of bolts
3	13½	7⅝	3½	7¾	¾	7½	6	4–⅝
3½	14⅞	8¾	4	9	¾	8½	7	4–⅝
4	16	9⅞	4½	10⅛	1	9	7½	8–⅝
5	16¾	12	5½	12⅞	1	10	8½	8–¾
6	18	15¼	6⅜	15¼	1	11	9½	8–¾
8	20¼	17½	8¼	19¾	1¼	13½	11¾	8–¾
10	22¼	21⅜	10⅛	25	1½	16	14¼	12–⅞
12	24	24¾	11⅞	29⅝	2	19	17	12–⅞
14	25¾	28⅜	13⅝	33⅝	2	21	18¾	12– 1
16	28	31⅞	15⅜	38¼	2½	23½	21¼	16– 1

Webster Low-pressure Receiver Oil Separators

These separators, acting as eliminators of oil and condensation and as receivers or mufflers, are used chiefly in exhaust steam lines between reciprocating engines and low or mixed-pressure turbines, or as receivers for the intermittent exhaust from groups of steam hammers.

They are of riveted steel construction, with cast-iron nozzles, and, like most of the Webster Oil Separators, are equipped with hooked steel multi-baffles. The nozzles are of cast iron with flanges drilled low-pressure standard.

The illustration shows one of the many forms of the Webster Low-pressure Receiver Oil Separator. The inlet and outlet nozzles may be located to conform with any direction of flow of steam. The axis of the shell may be either horizontal or vertical.

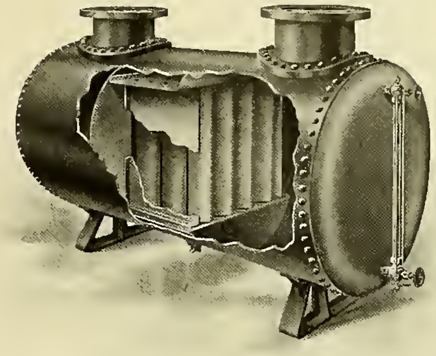


Fig. 24-31. The Webster Low-pressure Receiver Oil Separator

Inquiries regarding the Webster Low-pressure Receiver Oil Separators should be accompanied by a sketch showing the proposed location of and space available for the separator, the sizes and locations of inlet and outlet nozzles and the direction of flow. The inquiry should state the maximum amount of steam to be purified.

Webster Grease and Oil Traps

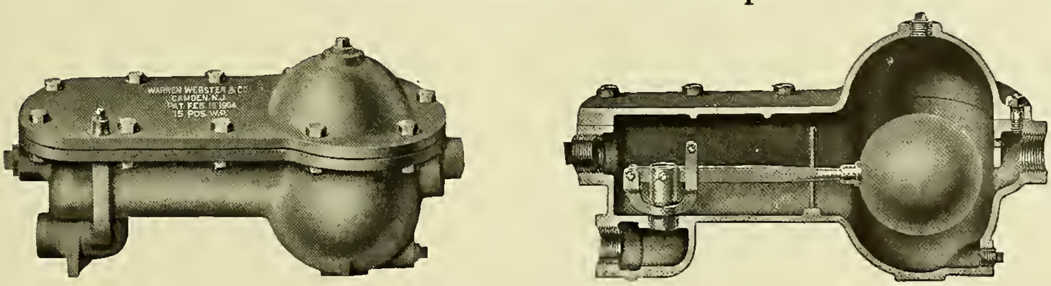


Fig. 24-32. The Webster Grease and Oil Trap

The Webster Grease Trap is for use in draining oil separators on exhaust steam lines or on feed-water heaters, or for removing from the course of the steam any accumulations of oily drips at other points in the low-pressure steam mains or branches. It will operate with equal efficiency under any pressure between atmospheric and 15 lb. per sq. in., above. It is not designed for use under high vacuum conditions.

As shown in the accompanying sectional illustration (Figure 24-32) the valve mechanism is simple. The discharge orifice is designed to give the full area of the inlet opening. The valve piece is conical and closes against a sharp-edged seat.

The ball float and valve chamber are easily reached for quick cleaning without disturbing pipe connections.

Properly installed, the Webster Grease Trap should be provided with a bypass in the piping around it; a check valve should be in the line beyond the outlet and bypass, and an equalizing or vent pipe should be run from the top of trap to the exhaust main beyond oil separator. See Figure 24-28.

Ratings for Webster Grease Traps: Because the mixture to be discharged is likely to be more or less viscous and sluggish in movement when it is cool it is impossible to rate grease traps on a condensation basis. The size of grease trap to be selected in any case should be that of the drip connection of the oil separator which it is to drain.

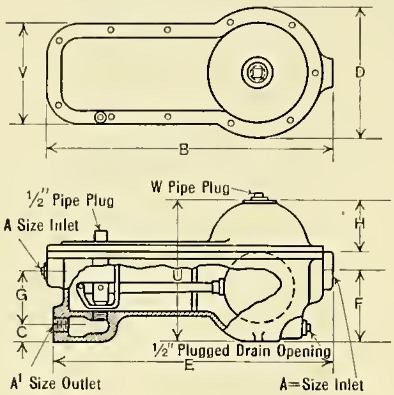


Fig. 24-33

Table 24-8. Dimensions of Webster Grease and Oil Traps

Number	A	A¹	B	C	D	E	F	G	H	U	V	W
016	¾	¾	15¾	1	8	15	4¼	3⅛	2⅞	8⅛	6¼	¾
116	1¼	1¼	19⅛	1⅜	9	18⅜	5⅜	3⅞	4	10¾	7	1
216	2	2	20⅝	1⅝	10½	19⅞	6⅛	4½	4⅜	12¼	8	1½

All dimensions in inches and subject to slight variation

The Webster Suction Strainer

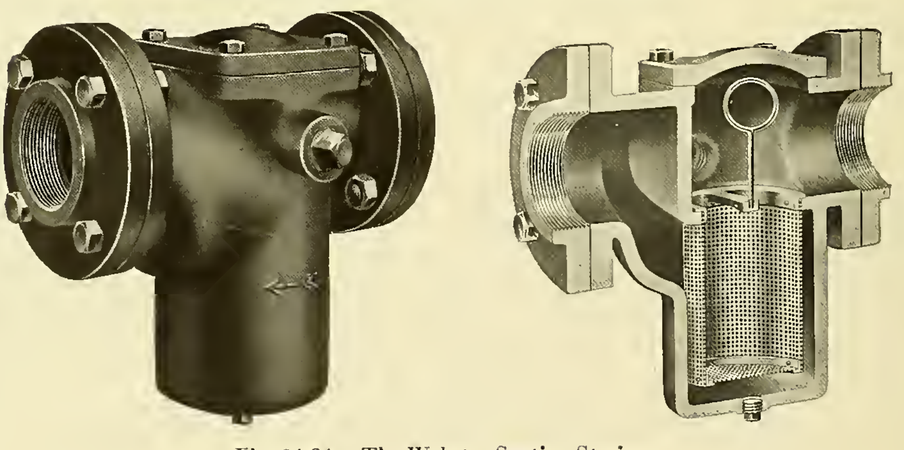


Fig. 24-34. The Webster Suction Strainer

The Webster Suction Strainer is used to prevent the passage to the vacuum pump of dirt and scale brought down with the condensation from a vacuum heating system. The use of this strainer prevents scoring of the pump-cylinder lining, valves and piston rods and the serious efficiency losses and repair bills that would follow such scoring. The strainer is provided with a tapping for the introduction of cold make-up water when same

is desired and when specially ordered, a spray nozzle is provided to insure thorough mixture of cold water and vapor in return. Another tapping is provided for a connection to the vacuum gauge and a third plugged outlet is for draining the body when the strainer is not in use. The shell and removable cover are of cast iron with composition gasket in the joint. Companion flanges, drilled low-pressure standard, are provided for inlet and outlet connections.

The basket is of perforated brass, and has at its top rim a casting in which is fastened a handle for lifting out the strainer. The perforations are 0.043 in. in diameter and of sufficient number to provide a total area twice that of the entering pipe.

The Webster Suction Strainer is to be placed in horizontal piping only, and should be set so that the axis of the body will be vertical. Water flows to it in the direction of the arrow (see Figure 24-34), and its course through the strainer is evident from the sectional view in the same figure.

During the cleaning process it is customary, if the system must be maintained in operation, to use either the relay pump or the ejector, if there is one, and if not, to temporarily run the returns by gravity to the sewer or waste, closing the stop-valve in the main return. The entire operation occupies but a few minutes.

Table 24-9. Dimensions of Webster Suction Strainer

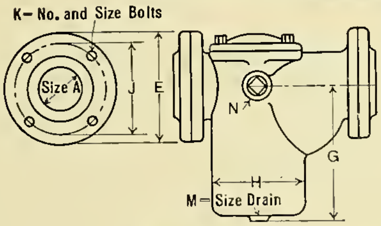
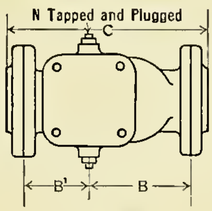


Fig. 24-35

For maximum working pressure of 15 lb. per sq. in.



Top view

All dimensions in inches and subject to slight variation

Size A	B	B¹	C	E	G	H	J	K	M	N
2	$5\frac{5}{8}$	$4\frac{3}{8}$	12	6	$6\frac{7}{8}$	$5\frac{1}{4}$	$4\frac{3}{4}$	$4\text{–}\frac{5}{8}\text{x}2$	$\frac{1}{2}$	$\frac{3}{4}$
3	$6\frac{5}{8}$	$4\frac{7}{8}$	$13\frac{3}{4}$	$7\frac{1}{2}$	$8\frac{7}{8}$	$5\frac{3}{4}$	6	$4\text{–}\frac{5}{8}\text{x}2\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{4}$
4	$8\frac{3}{16}$	$5\frac{13}{16}$	$16\frac{3}{8}$	9	$10\frac{5}{8}$	$7\frac{1}{8}$	$7\frac{1}{2}$	$8\text{–}\frac{5}{8}\text{x}2\frac{3}{4}$	$\frac{1}{2}$	$\frac{3}{4}$
5	$9\frac{5}{8}$	$6\frac{3}{8}$	$18\frac{5}{8}$	10	$12\frac{1}{8}$	$8\frac{1}{4}$	$8\frac{1}{2}$	$8\text{–}\frac{3}{4}\text{x}2\frac{3}{4}$	$\frac{1}{2}$	$\frac{3}{4}$
6	$10\frac{15}{16}$	$7\frac{1}{16}$	$20\frac{7}{8}$	11	$13\frac{11}{16}$	$9\frac{5}{8}$	$9\frac{1}{2}$	$8\text{–}\frac{3}{4}\text{x}2\frac{3}{4}$	$\frac{1}{2}$	$\frac{3}{4}$
7	$12\frac{9}{16}$	$9\frac{7}{16}$	25	$12\frac{1}{2}$	$19\frac{1}{4}$	13	$10\frac{3}{4}$	$8\text{–}\frac{3}{4}\text{x}3$	$\frac{3}{4}$	1
8	$14\frac{1}{8}$	$9\frac{7}{8}$	$27\frac{1}{4}$	$13\frac{1}{2}$	21	$14\frac{1}{2}$	$11\frac{3}{4}$	$8\text{–}\frac{3}{4}\text{x}3\frac{1}{8}$	$\frac{3}{4}$	1
10	$17\frac{1}{4}$	$11\frac{1}{4}$	$32\frac{1}{4}$	16	$24\frac{7}{8}$	$16\frac{3}{4}$	$14\frac{1}{4}$	$12\text{–}\frac{7}{8}\text{x}3\frac{1}{2}$	$\frac{3}{4}$	1
12	21	$12\frac{7}{8}$	38	19	29	20	17	$12\text{–}\frac{7}{8}\text{x}3\frac{1}{2}$	$\frac{3}{4}$	1

Webster Dirt Strainers

Webster Dirt Strainers are used in steam heating systems to prevent dirt from entering radiator traps or traps on drip points, mains or blast coils. They provide convenient receptacles for retention and accumulation of pipe chips, rust, dirt, etc., where impurities can do no harm and where they are easily and quickly removed.

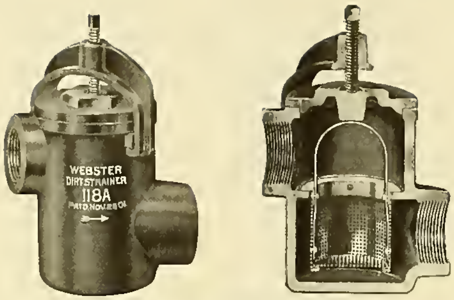


Fig. 24-36. Class A (Offset)

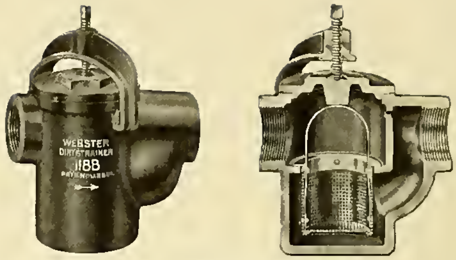


Fig. 24-37. Class B (Straightway)

Webster Dirt Strainers

Two models are made: Class A with offset and Class B with straightway pipe connections. Both have cast-iron shell and cover, the latter made easily removable by means of a yoke and screw.

The basket is made from sheet brass perforated with 0.043-in. diameter holes. The total free area through the basket is several times the area of the entering pipe. The sides of the basket are reinforced with strips which are continued upward to form a bale handle. This handle not only serves to make the basket easily removable but acts as a spring against the cover to hold the basket in place.

The range of types and sizes offers a selection for any service conditions.

The use of these strainers greatly lessens the amount of attention required to keep the system in thoroughly efficient operation and eliminates incentive for the neglect always to be expected with dirt pockets composed of pipe fittings, which cost nearly as much to make and are never as good.

Table 24-10. Dimensions of Webster Dirt Strainers, Classes A and B

Maximum pressure, 15 lb. per sq. in.

Dimensions in inches and subject to slight variation

Class A

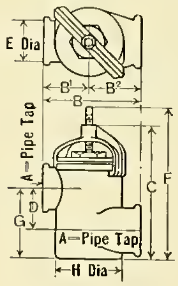


Fig. 24-38

Class B

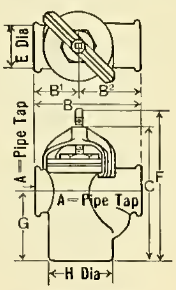


Fig. 24-39

Class A.—Offset (Fig. 24-38)

No.	Size A	B	B¹	B²	C	D	E	F	G	H
018-A	½ or ¾	3½	1¾	1¾	5 5/16	1¾	1⅝	6	2¾	2½
118-A	1 or 1¼	4¾	2¼	2½	6⅝	2	2⅛	7 5/16	3½	3¼
218-A	1½ or 2	6	2¾	3¼	8 11/16	3	3	9 3/16	4¾	4¼

Class B.—Straightway (Fig. 24-39)

No.	Size A	B	B¹	B²	C	E	F	G	H
018-B	½ or ¾	4¼	1¾	2½	5 5/16	1⅝	6	2¾	2½
118-B	1 or 1¼	5½	2¼	3¼	6⅝	2⅛	7 5/16	3½	3¼
218-B	1½ or 2	7¼	2¾	4½	8 11/16	3	9 3/16	4¾	4¼

The Webster Vacuum-pump Governor

The vacuum pump of a vacuum heating system should be as nearly automatic in operation as possible.

The Webster Vacuum-pump Governor automatically controls the admission of steam to the pump cylinder or cylinders in proportion to the degree of vacuum required. When only part of the heating load is on, just enough

steam is admitted into the pump to produce the degree of vacuum required. When the need is greater, the supply of steam is automatically increased.

The Webster Vacuum-pump Governor can be adjusted to control the vacuum to any predetermined degree, and may be readjusted when necessary. It is remarkably sensitive through a wide range of adjustment.

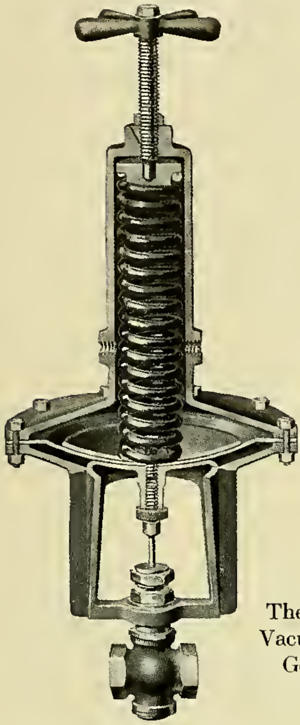


Fig. 24-40

The Webster Vacuum-pump Governor

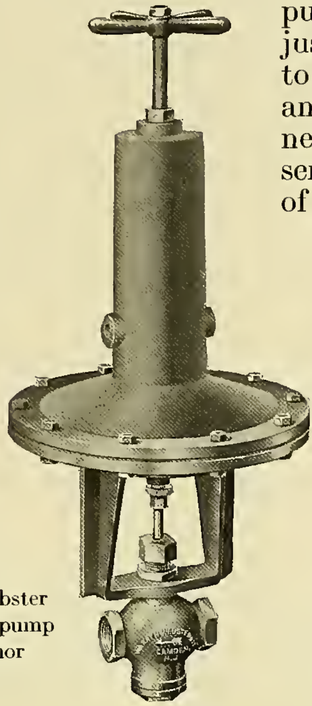


Fig. 24-41

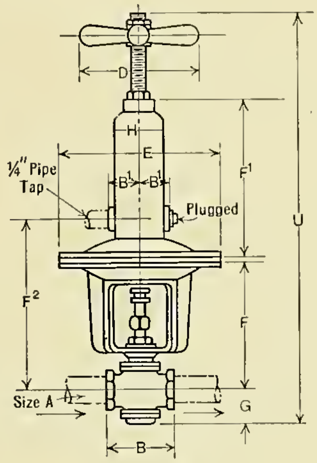


Fig. 24-42

Table 24-11. Dimensions of Webster Vacuum-pump Governors

Size A	B	B¹	D	E	F	F¹	F²	G	H	U
¾	2⅞	1¾	5	9⅞	7½	10¼	10⅝	$1\frac{9}{16}$	2⅞	$23\frac{11}{16}$
1	3⅜	1¾	5	9⅞	7⅞	10¼	11	2	2⅞	24½
1¼	4	1¾	5	9⅞	8	10¼	11⅛	2	2⅞	24⅝
1½	4½	1¾	5	9⅞	$8\frac{9}{16}$	10¼	$11\frac{11}{16}$	$2\frac{5}{16}$	2⅞	25½
2	5½	1¾	5	9⅞	8⅞	10¼	12	$2\frac{9}{16}$	2⅞	$26\frac{1}{16}$
2½	6⅜	1¾	5	9⅞	10¾	10¼	13⅞	3¼	2⅞	28½
3	7⅛	1¾	5	9⅞	11⅛	10¼	14¼	3⅞	2⅞	29½
3½	8	1¾	5	9⅞	11½	10¼	14⅝	4	2⅞	30

The Webster Suction Strainer and Vapor Economizer

This special device, in addition to its function of protecting the vacuum pump, has a particular advantage in vacuum heating systems where some unusual operating condition results in the return of water to the vacuum pump at a high temperature.

Under such conditions, re-evaporation or transformation of water into steam vapor may occur, and the presence of this steam vapor adds to the duty of and may interfere with the proper operation of the pump.

If cold water is constantly required for making up the boiler-feed water it can be introduced in the standard Webster Suction Strainer, by the use

of the Webster spray-head, without increasing the cost of plant operation. The special Webster Suction Strainer and Vapor Economizer is designed to meet conditions where cooling water is required, but where the use of it as make-up water would entail waste.

The cold water is passed around a nest of copper coils and absorbs the heat of the steam vapor in the main return.

This water is not handled by the vacuum pump and does not mix with the condensation in the main return line, as the economizer becomes merely an extension of the hot-water piping system, under the available pressure.

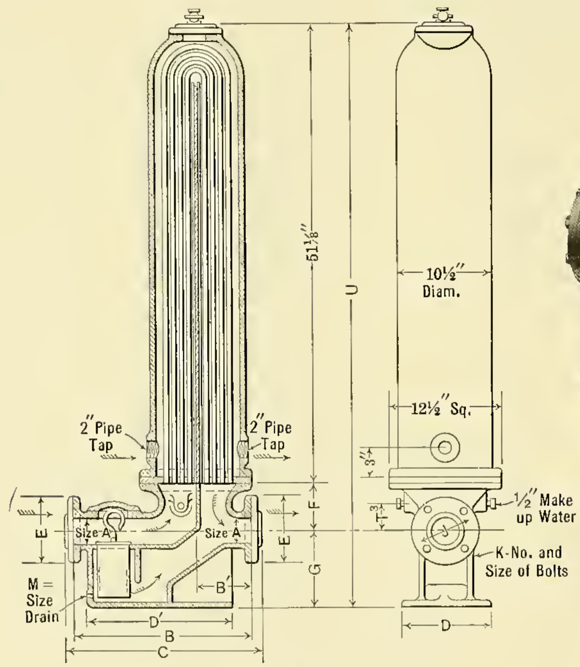


Fig. 24-43

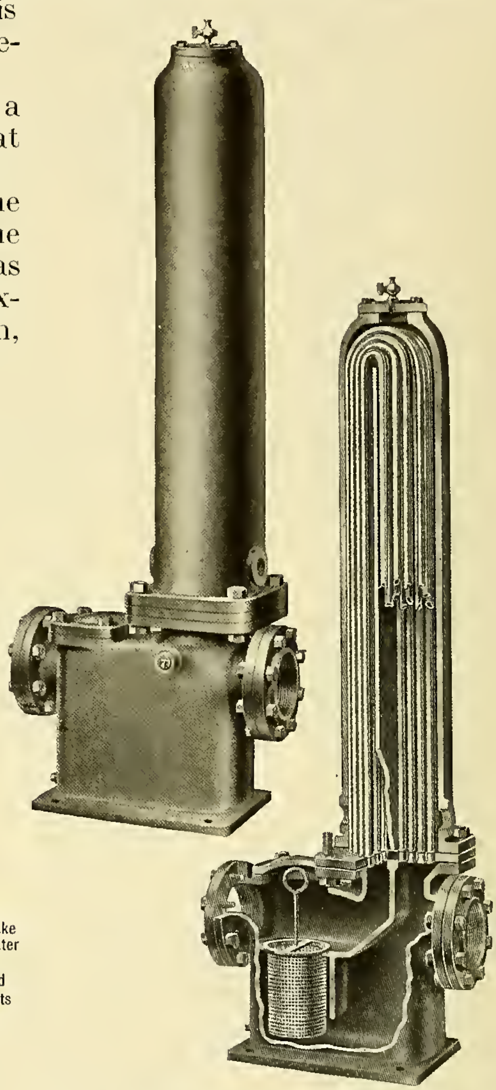


Fig. 24-44. The Webster Suction Strainer and Vapor Economizer

Table 24-12. Dimensions of the Webster Suction Strainer and Vapor Economizer

All dimensions in inches and subject to slight variation

Size A	B	B[1]	C	D	D[1]	E	F	G	J	K	T[3]	U	M
3	19⅝	6	22	10	15¾	7½	5½	8⅝	6	4–⅝ x 2¼	2¾	65¼	½
5	22¾	6¼	25½	12½	18⅞	10	6⅜	12	8½	8–¾ x 2¾	3½	69½	½
7	28¼	7¼	31⅜	14½	24¼	12½	7⅞	19	10¾	8–¾ x 3	4⅜	78	¾

Webster Lift Fittings—Series 20

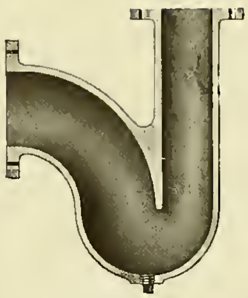


Fig. 24-45.
Webster Lift Fitting

Webster Lift Fittings are special devices used in pairs at points in vacuum heating systems where condensation is to be lifted to a higher level. The condensation is lifted vertically to a higher level in "slugs" on the air-lift principle; the slugs being obtained by the use of a comparatively small diameter vertical return with its lower end submerged in the well below the level of the horizontal return which it drains. The lower lift fitting allows the condensation to accumulate in the well below the inlet connection until it seals the vertical passage, thus causing a slight reduction of the vacuum on the inlet side and forcing the water from the well through the vertical lift pipe to the higher level. The upper lift fitting allows the condensation to flow into the horizontal return without falling back into the lifting line.

Lifts of six feet or over should be made in steps rather than all in one rise. Steps should be used instead of "drag" lifts through long upwardly inclined pipes. In any case, the pipes between lifts must grade toward pump.

Webster Lift Fittings are a big improvement upon and should be substituted for the home-made fittings which in the past have had to be made from combinations of ordinary tees or crosses and plugs, because nothing better was obtainable. Each Webster Lift Fitting is a unit casting, neat in appearance and correctly proportioned for capacity of well and for the area ratio of inlet to outlet. The use of these fittings eliminates all the guesswork and uncertainty about proper operation. They cost less than combinations of fittings when the labor cost as well as that of the fittings is considered.

Each fitting is provided with a clean-out plug for removing any accu-

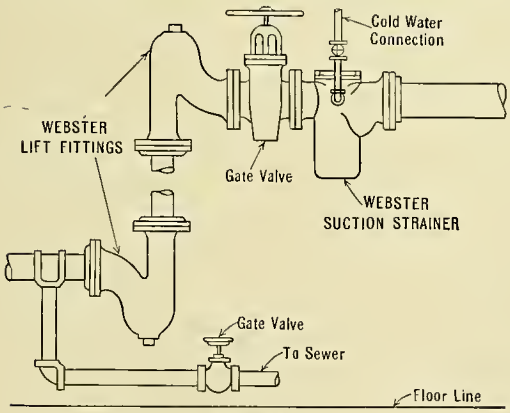


Fig. 24-46.
Typical application of Webster Lift Fittings
(See also Fig. 13-1, page 139)

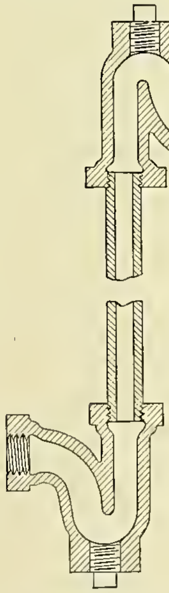


Fig. 24-47.
Long screwed
lift connection

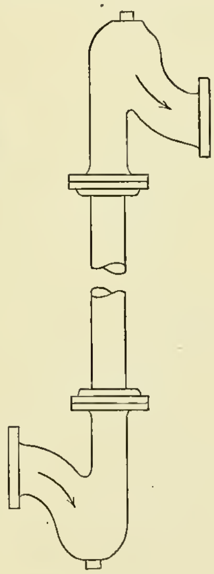


Fig. 24-48.
Long flanged
lift connection

mulation of dirt or other foreign matter from the lift pocket. The larger sizes are flanged and finished and drilled to the low-pressure standard.

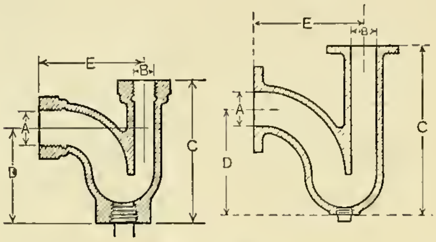


Fig. 24-49

Table 24-13. Dimensions of Series 20 Webster Lift Fittings in inches

Size		A	B	C	D	E	Drain
		Inlet	Outlet				
3/4	Screwed	3/4	1/2	3 7/8	2 5/8	2 3/4	1/2
1	"	1	3/4	4 3/8	3	3 1/8	1/2
1 1/4	"	1 1/4	1	5 1/4	3 1/2	3 3/4	3/4
1 1/2	"	1 1/2	1	6 1/16	4 1/16	4 1/4	3/4
2	"	2	1 1/4	6 7/8	4 5/8	4 7/8	1
2 1/2	"	2 1/2	1 1/2	8 1/8	5 3/8	5 7/8	1
3	Flanged	3	2 1/2	14 1/4	9	9 3/8	1
4	"	4	3	17 1/8	10 5/8	11 1/4	1
5	"	5	3 1/2	19 3/4	12 1/2	12 15/16	1
6	"	6	4	21 3/4	13 7/8	14 5/16	1
8	"	8	4 1/2	25 1/4	16 1/4	17	1
10	"	10	6	31 1/8	20 1/8	20 5/8	1
12	"	12	7	34 1/2	22 5/8	23 3/16	1

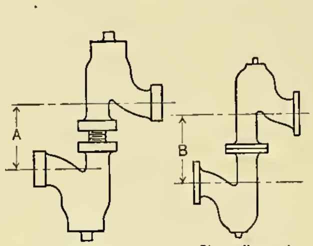


Fig. 24-50

Table 24-14. Minimum Distance Between Centers

3/4-in. screwed fittings			A = 3 1/8 in.
1-in.	"	"	A = 3 1/4 in.
1 1/4-in.	"	"	A = 4 1/8 in.
1 1/2-in.	"	"	A = 4 3/4 in.
2-in.	"	"	A = 5 1/2 in.
2 1/2-in.	"	"	A = 8 in.
3-in. flanged fittings			B = 10 9/16 in.
4-in.	"	"	B = 13 1/16 in.
5-in.	"	"	B = 14 9/16 in.
6-in.	"	"	B = 15 13/16 in.
8-in.	"	"	B = 18 1/16 in.
10-in.	"	"	B = 22 1/16 in.
12-in.	"	"	B = 23 13/16 in.

Webster Receiving Tanks—Plain, Water-control and Steam-control Types

These tanks are used in connection with vacuum steam heating systems, to provide a place for storage of the condensation discharged by the vacuum pump and for liberation of the air that comes over with this condensation. Each type is designed for pressures not exceeding 30 lb. per sq. in., for installation in horizontal position, and each type has proper receiving capacity and air-liberating surface.

The Plain Type receives the condensation and air through an end opening near the top. The air escapes through a vent in the top of the tank, and the water flows by gravity to the bottom outlet and to the feed-water heater or other point of disposal. If the rate of flow of returns to tank exceeds rate of discharge from tank, the excess overflows through an opening on the end near the top.

The Water-control and Steam-control Types have regulating valves which are operated by sink pan and rigging similar to those used to regulate the water level in Webster Feed-water Heaters. These two types are also provided with water-troughs, to insure best operation of the sink pan.

The Water-control Type has its regulating valve arranged to automatically admit "make up" at all times when the returns from the heating system are temporarily insufficient to keep the water level in the tank at the pre-

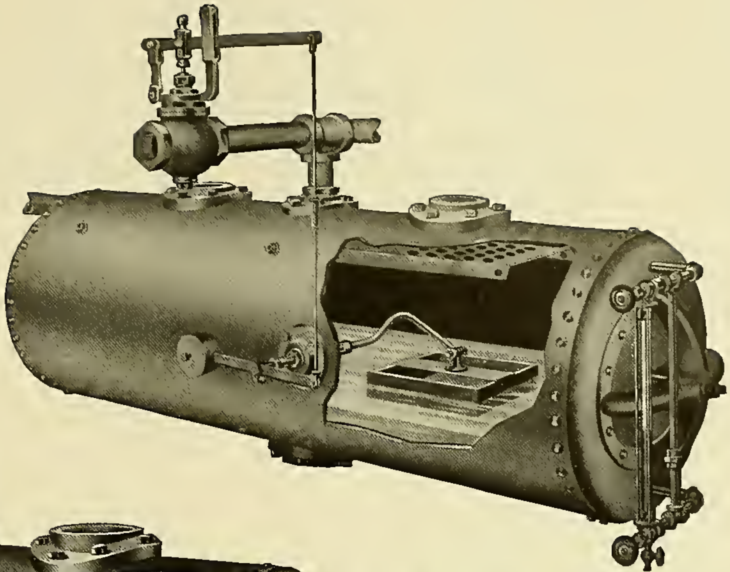


Fig. 24-52. Webster Air-separating Tank and Receiver, Steam-control Type

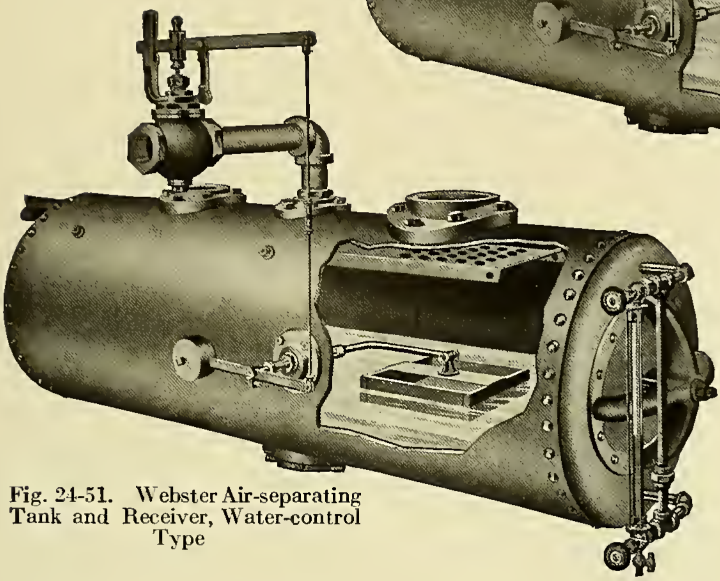


Fig. 24-51. Webster Air-separating Tank and Receiver, Water-control Type

determined point. The air is vented to atmosphere, the water flows by gravity to the heater or other place of disposal, and any excess of water overflows, as with the Plain Type.

The Steam-control Type, which is used where the boiler or boilers are to be fed in proportion to the returns reaching the receiving tank, has its regulating valve installed in the steam supply line to the boiler-feed pump. With water in the tank at or above the predetermined level, the boiler-feed pump is in operation, feeding the returns into the boiler, but when the tank level is below normal, the steam to the boiler-feed pump is shut off and the pump stopped until sufficient returns collect again. Make-up water, if necessary, may be introduced into the tank by hand. The venting of air to atmosphere, delivery of water by gravity flow and provision for overflow of excess water are the same as in the Plain Type.

All three types of Webster Receiving Tanks are made from riveted flange steel and have flat heads. The Water-control and Steam-control Types have removable manhole covers and gauge fittings in one end. Each tank is hand-made throughout from best obtainable materials. The sizes listed are standard. Larger sizes can be made upon special order.

Table 24-15. Dimensions of Webster Receiving Tanks

Note: Openings will be bushed to suit requirements. All dimensions in inches

Plain Type

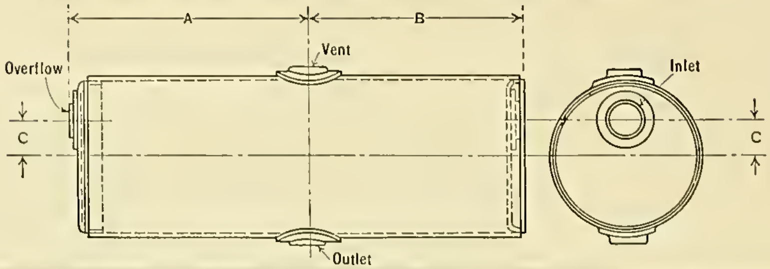


Fig. 24-53

Size	Inlet	Outlet	Air vent	Overflow	A	B	C
18 x 48	4	4	1½	4	25½	25	3½
24 x 72	5	5	2	5	38½	37	6
36 x 96	8	8	3	6	50¼	49	11½

Water-control Type

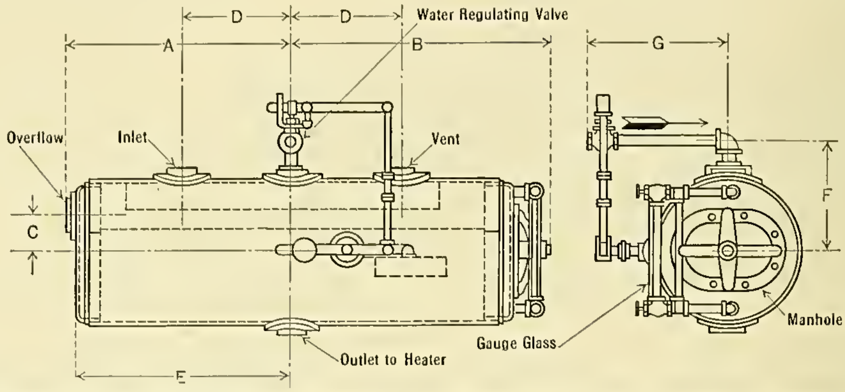


Fig. 24-54

Size	Inlet	Outlet	Air vent	Overflow	Reg. valve	A	B	C	D	E	F	G
18 x 48	4	4	1½	4	1	25½	30½	3½	12	24¼	14	18¼
24 x 72	5	5	2	5	1½	37⅝	42½	6	18	36¼	18	23
36 x 96	8	8	3	6	2	50¼	54⅝	11½	18	48⅜	24	28¼

Steam-control Type

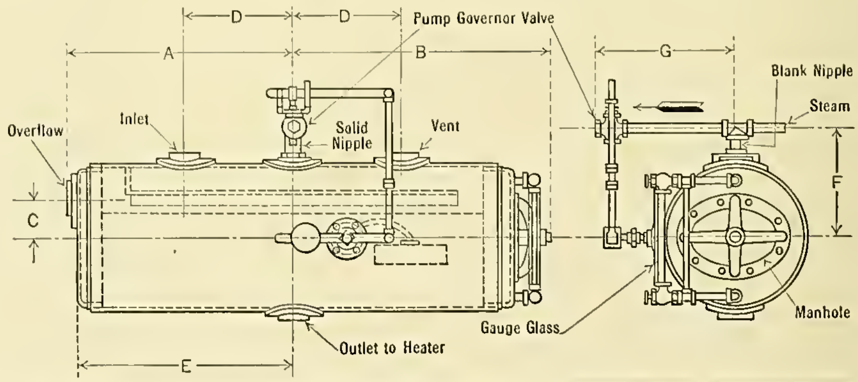


Fig. 24-55

Size	Inlet	Outlet	Air vent	Overflow	Gov. valve	A	B	C	D	E	F	G
18 x 48	4	4	1½	4	1	25½	30½	3½	12	24¼	14	18¼
24 x 72	5	5	2	5	1½	37⅝	42½	6	18	36¼	18	23
36 x 96	8	8	3	6	2	50¼	54⅝	11½	18	48⅜	24	28¼

For ratings see Table 13-1, page 138

The Webster Water Accumulator

Fig. 24-56. The Webster Water Accumulator

This is a cast-iron fitting of oval cross section, designed to accumulate condensation for the protection of the diaphragms of pressure-reducing

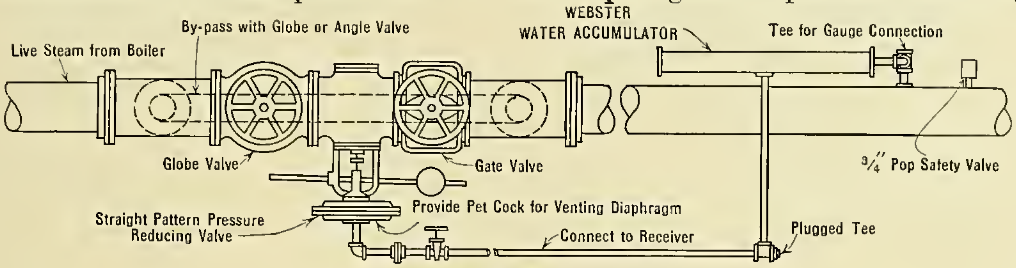


Fig. 24-57

valves and similar apparatus against the heat of steam which would deteriorate the diaphragms if brought into direct contact. This application is shown in Figure 24-57.

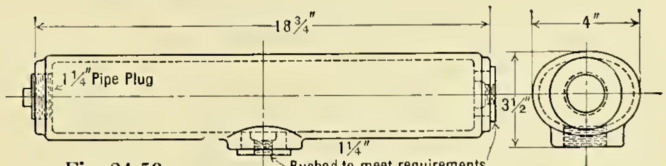


Fig. 24-58

The Webster Water Accumulator may also be used to provide protection for low-pressure steam gauges.

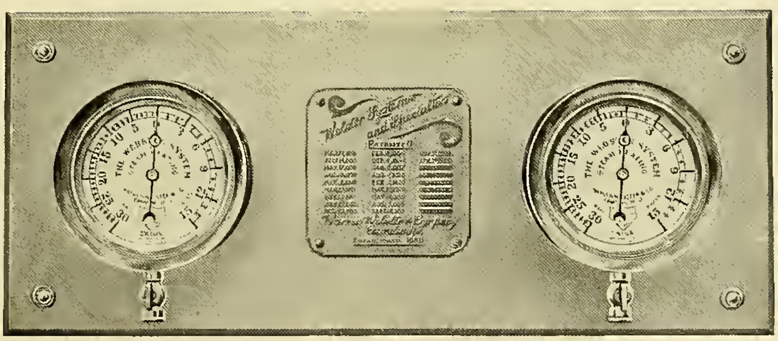


Fig. 24-59. Webster Combination Gauges

Gauges for Webster Systems

Webster Gauges are of high quality and are furnished in various standard forms, and to suit special specifications. The usual outfit furnished with Webster Vacuum Systems is a set of two 5½-in. face, nickel-plated combination pressure and vacuum gauges, mounted on Monson, Me., slate board with Webster System name plate.

Single combination gauges can be furnished, either for Vacuum or Modulation Systems, in 5½-in. size.

Single gauges are also furnished with Webster Hylo Vacuum Sets, as elsewhere described.

Larger gauges, slate or marble boards for three or four gauges, or gauges having special graduations or marking can also be furnished when required.

The Webster Modulation Vent Trap

This device is installed in the low point of the dry-return line of the Webster Modulation System before the returns flow to the boiler or boilers as feed

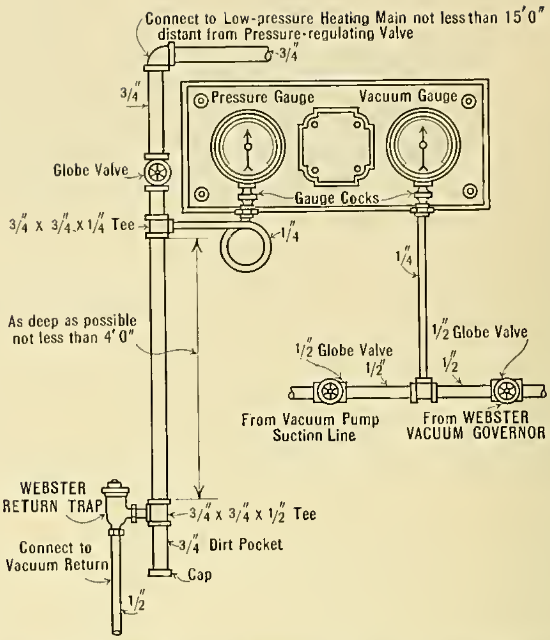


Fig. 24-60. Connections for gauges, Webster Vacuum System

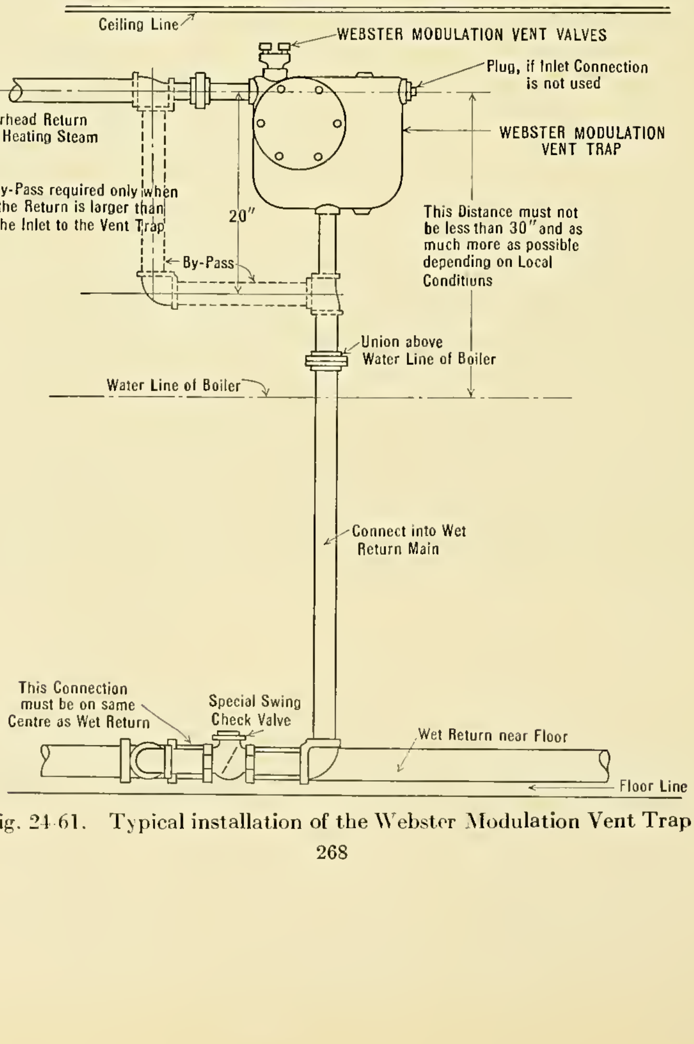


Fig. 24-61. Typical installation of the Webster Modulation Vent Trap

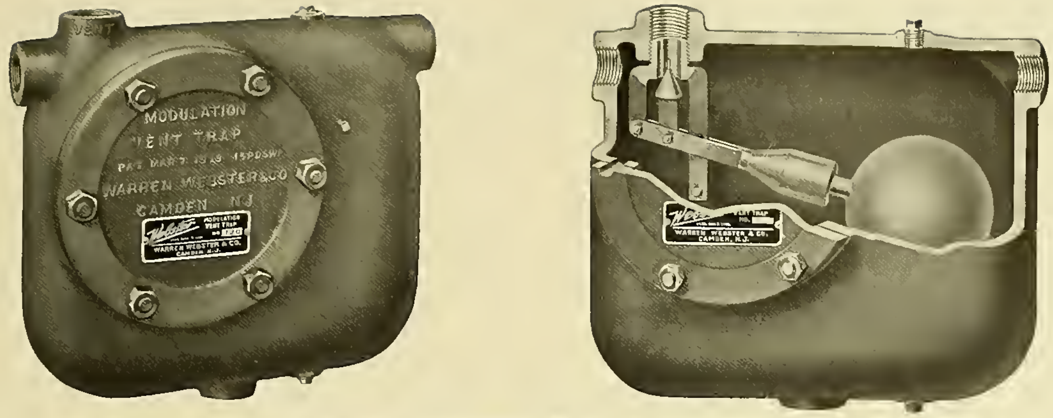


Fig. 24-62. The Webster Modulation Vent Trap

water. It affords a simple, dependable method of venting the entrained air to atmosphere and of automatically insuring the return of the water to the boiler under fluctuating boiler pressures. The air vent is controlled by an internal float mechanism. The valve piece is conical and closes against a sharp-edged seat.

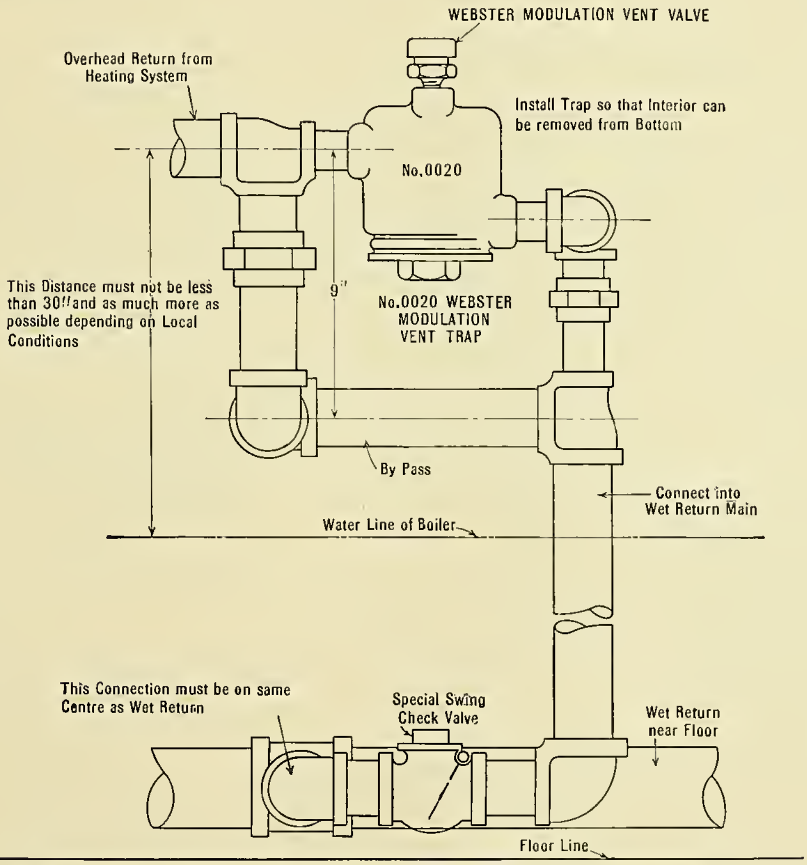


Fig. 24-63. Typical installation of Webster Modulation Vent Trap No. 0020

Other means for returning water to the boiler are provided for unusual structural features of the building or conditions of use, but for the average building to which the Webster Modulation System is adaptable the Webster Modulation Vent Trap is used.

In the illustrations, Figures 24-61 and 24-63, Webster Modulation Vent Valves are shown in position at the air outlets of the Vent Traps. These valves are always required where it is desired to circulate steam below atmospheric pressure at intervals. Where large hot-water generators are used, or where a part or all of the radiators are under automatic control, the vent valves should be omitted unless vacuum breakers are provided on the return lines at the proper places.

The type of Modulation Vent Trap shown in Figures 24-61, 24-62 and 24-64 is that which is used for the larger systems. For installations such as small residences, the size 0020 Trap as shown in Figures 24-63 and 24-65 is most often used. Capacity ratings are given on page 240.

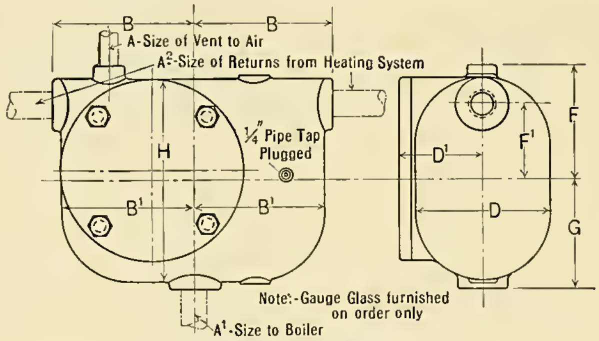


Fig. 24-64. Dimensions of Webster Modulation Vent Trap, Series 20 (See Table 24-16)

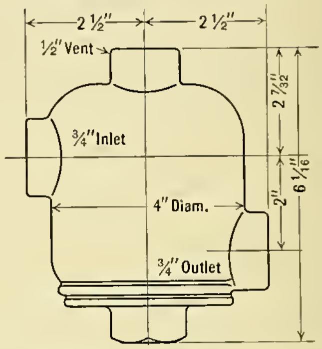


Fig. 24-65. Dimensions of Webster Modulation Vent Trap Number 0020

Table 24-16. Dimensions of Webster Modulation Vent Traps, Series 20, Fig. 24-64

SIZE	A	A¹	A²	B	B¹	D	D¹	F	F¹	G	H
No. 020	$\frac{1}{2}$	1	1	$6\frac{7}{8}$	$6\frac{1}{2}$	$6\frac{3}{4}$	$4\frac{1}{8}$	$5\frac{5}{8}$	$3\frac{3}{4}$	$5\frac{3}{8}$	10
No. 120	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$9\frac{3}{16}$	$8\frac{1}{16}$	8	$4\frac{7}{8}$	$8\frac{1}{2}$	$6\frac{1}{4}$	$8\frac{1}{4}$	$15\frac{1}{2}$
No. 220	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$9\frac{3}{16}$	$8\frac{11}{16}$	8	$4\frac{7}{8}$	$8\frac{1}{2}$	$6\frac{1}{4}$	$8\frac{1}{4}$	$15\frac{1}{2}$
No. 320	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$9\frac{3}{16}$	$8\frac{11}{16}$	8	$4\frac{7}{8}$	$8\frac{1}{2}$	$6\frac{1}{4}$	$8\frac{1}{4}$	$15\frac{1}{2}$

The Webster Modulation Vent Valve

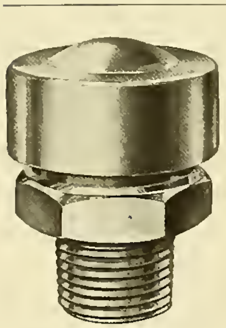


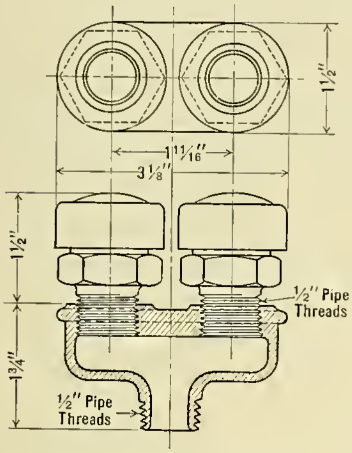
Fig. 24-66. Webster Modulation Vent Valve

This valve has been specially devised to meet the requirement for check against inflow of air to a modulation system when it is desired to operate at a pressure less than atmospheric. This check is provided by the seating of a hollow seamless ball which is retained by a cage structure as shown in Figure 24-66.

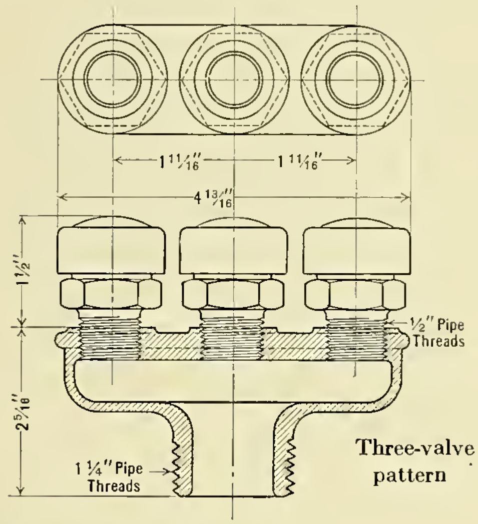
Due to the very slight weight of the ball and the construction of the valve body and seat, a pressure less than one ounce per square inch will serve to lift the valve from its seat, thus permitting the escape of air from the Vent Trap.

The Modulation Vent Valve is made in only the ½-in. size which

may be used as a single unit for installations up to 8500 sq. ft. of direct radiation or equivalent. For larger installations these valves are furnished in multiple units of the necessary number with a fitting such as that shown in Figure 24-67. See Table 23-9, page 240.



For use where two vent valves are required



Three-valve pattern

Fig. 24-67. Multiple-unit Webster Modulation Vent Valves

The Webster Damper Regulator

The Webster Damper Regulator is used with the Webster Modulation System and automatically controls the opening of the draft door and check damper of the low-pressure steam-heating boiler. It is extremely sensitive and accurate because of the ample diaphragm area and controls the fire to maintain the steam pressure always within a few ounces of that for which the regulator is set.

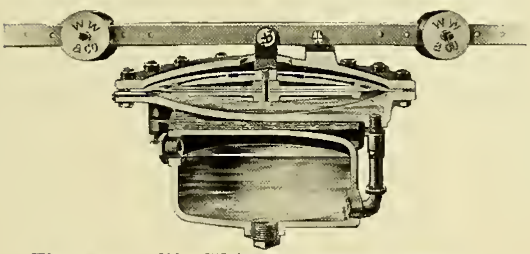


Fig. 24-68. The Webster Damper Regulator

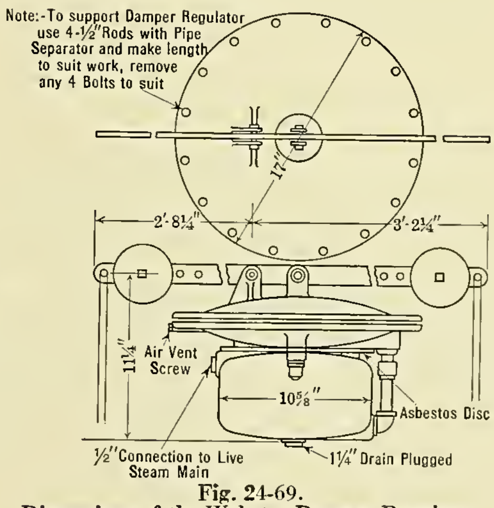


Fig. 24-69. Dimensions of the Webster Damper Regulator

Table 24-17. Power Developed by Webster Damper Regulator

The following figures, based upon tests with lever in mid-position, afford a comparison with other damper regulators having much smaller diaphragms

Pressure in lb. per sq. in.	0.5	1.0	2.0	3.0	4.0	5.0
Average pull at end of lever, lb.	4.125	8.25	16.5	24.75	33.0	41.25

Webster Hylo Vacuum-control Sets

Each Webster Hylo Set consists of a Webster Hylo Vacuum Controller, handling vapor and air only, a Webster Hylo Trap, handling water of condensation only, Webster Hylo Vacuum Gauges, and when needed, Webster Lift Fittings.

The Webster Hylo Vacuum Controller regulates the vacuum from the low to the high vacuum through the action of the diaphragm and pilot valve. The vacuum differential, as fixed by the position of the weights on the diaphragm lever, may be adjusted to maintain the desired vacuum.

The Webster Hylo Trap permits condensation to flow from low to high vacuum without loss of differential. This trap is of ball-float type, with outlet water sealed.

The Webster Vacuum Gauges indicate the vacuum conditions upon both sides of the controller. Special arrangements of gauges and boards are furnished for varying requirements.

Webster Lift Fittings are required where returns must be lifted such as the case shown in Figure 15-7, page 177.

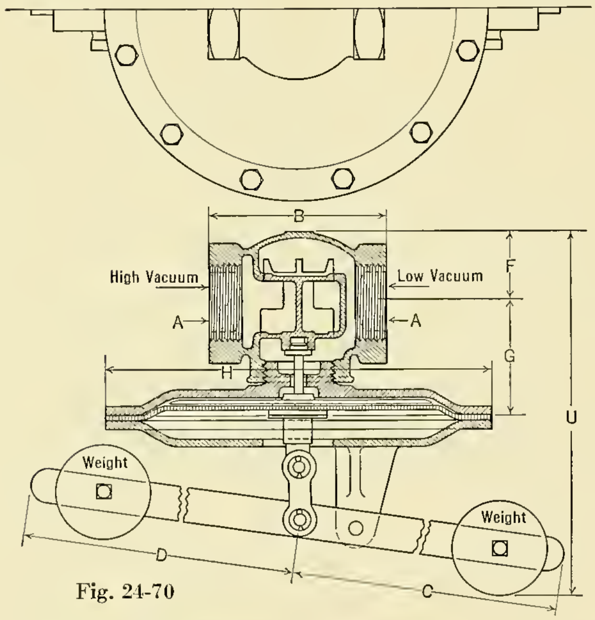


Fig. 24-70

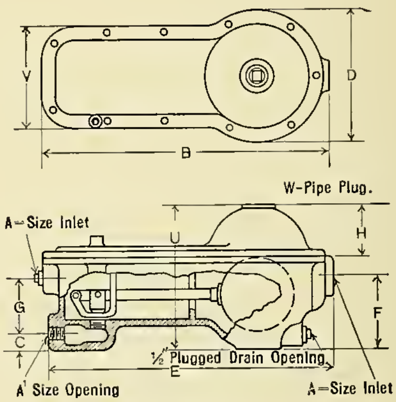


Fig. 24-71

Table 24-18. Dimensions of Webster Hylo Controller

All dimensions in inches and subject to slight variation

Size A	B	C	D	F	G	H	U
1	3½	12¾	11¼	1¼	2¼	10¾	9¼
1½	4¼	12¾	11¼	1¾	2⅞	10¾	10
2	5	12¾	11¼	2	3	10¾	10½

Table 24-19. Dimensions of Webster Hylo Traps for 15-lb. Working Pressure

All dimensions in inches and subject to slight variation

Number	A	A'	B	C	D	E	F	G	H	U	V	W
016	¾	¾	15¾	1	8	15	4⅛	3⅛	2⅞	8⅛	6¼	¾
116	1¼	1¼	19⅛	1⅜	9	18⅜	5⅜	3⅞	4⅛	10¾	7	1
216	2	2	20⅝	1⅝	10½	19⅞	6⅛	4½	4⅜	12¼	8	1½

The ratings are the same as for Webster Heavy-duty Traps, as given in Table 23-80, page 239

The Webster Sylphon Conserving Valve

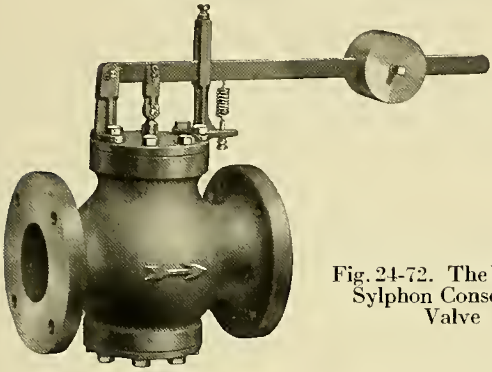
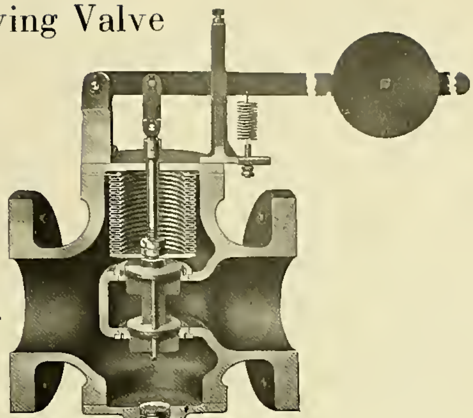


Fig. 24-72. The Webster Sylphon Conserving Valve



This valve is one of the special devices used in connection with the Webster Conserving System where steam is furnished direct from low-pressure heating boilers which are required to supply steam for other purposes than warming the building, at a constant pressure above that required for the heating system alone. It also insures the constant operation of the low-pressure steam-driven vacuum pump.

It is placed in the main steam line from boiler, the steam connection to vacuum pump being taken from the inlet side of the conserving valve. The pressure for which the conserving valve is set must be built up on the inlet side, before the conserving valve will open and allow steam to enter the low-pressure heating main.

In consequence, the vacuum pump will automatically start into operation before steam is admitted into the low-pressure heating main. The partial vacuum created in the return mains and radiators assures quick circulation as soon as the conserving valve automatically opens and permits the steam to flow into the main.

When steam supply is cut off from the heating system the pump will continue to operate until the condensation is thoroughly drained, assuring the return of all of the condensation to the boiler. With the types of boiler used with heating systems of this design, this is a very important matter. See pages 173 to 176.

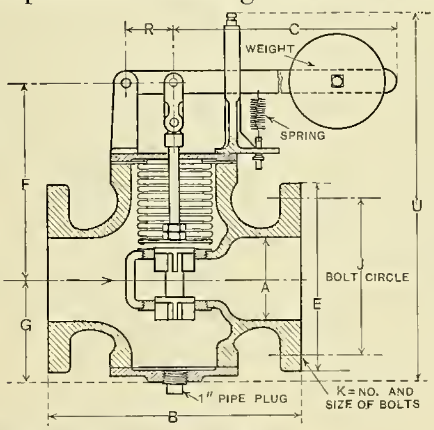


Fig. 24-73

Table 24-20. Dimensions of Webster Sylphon Conserving Valves

All dimensions in inches and subject to slight variation

Size A	B	C	E	F	G	J	K	R	U
4	12	20	9	$9\frac{3}{8}$	$4\frac{7}{8}$	$7\frac{1}{2}$	8—$\frac{5}{8}$	$2\frac{1}{4}$	$17\frac{1}{2}$
5	12	20	10	$9\frac{5}{8}$	$5\frac{1}{4}$	$8\frac{1}{2}$	8—$\frac{3}{4}$	$2\frac{1}{4}$	$18\frac{1}{8}$
6	13	$31\frac{1}{4}$	11	$10\frac{1}{4}$	$6\frac{1}{16}$	$9\frac{1}{2}$	8—$\frac{3}{4}$	$2\frac{3}{4}$	$21\frac{5}{8}$
8	$13\frac{1}{4}$	$31\frac{1}{4}$	$13\frac{1}{2}$	$11\frac{1}{4}$	$7\frac{3}{16}$	$11\frac{3}{4}$	8—$\frac{3}{4}$	$2\frac{3}{4}$	$24\frac{3}{4}$
10	15	$36\frac{9}{16}$	16	$12\frac{1}{2}$	$8\frac{3}{8}$	$14\frac{1}{4}$	12—$\frac{7}{8}$	$3\frac{7}{16}$	$28\frac{5}{8}$

The Webster Low-pressure Boiler Feeder—Series 16

In connection with heating boilers fed from hydro-pneumatic tanks, and under certain other conditions, a Webster Boiler Feeder is useful. This device is shown on Page 147, as part of a Webster Hydro-pneumatic System.

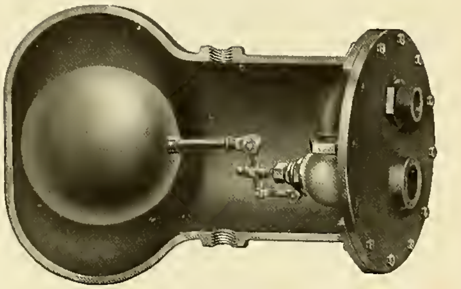


Fig. 24-71. Webster Low-pressure Boiler Feeder

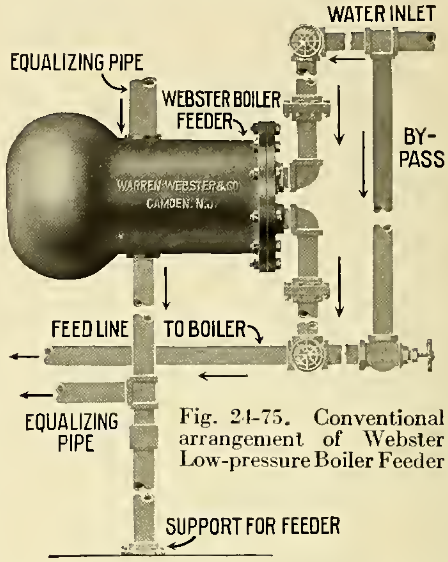


Fig. 24-75. Conventional arrangement of Webster Low-pressure Boiler Feeder

When the water level in the boiler lowers, the ball float opens the feed valve and allows the water to discharge directly to boiler.

The valve is of the double-balanced type with large orifice area, because of the low differential between the tank pressure and the boiler pressure. The ball float is large enough to give the power required to move the valve lever without excessive difference of water level.

An important point in the construction of the boiler feeder is that the valve and gear are within the casing. There are no outside glands to keep tight and any leakage which occurs is within the body of the device and hence into the boiler.

The working parts are easily accessible, but seldom need attention.

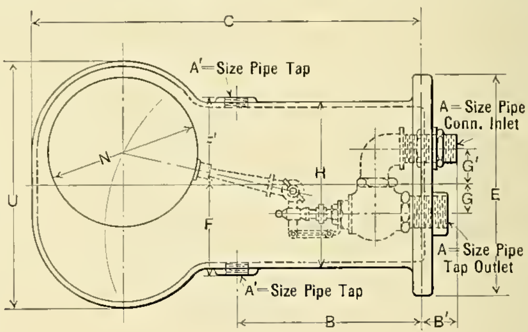


Fig. 24-76

Table 24-21. Dimensions of Series 16 Webster Low-pressure Boiler Feeder
Dimensions in inches and subject to slight variation

Number	A	A^{1}	B^{1}	B^{1}	C	E	F	G	G^{1}	H	N	U
116	½	1	12½	2	25⅜	14½	6¼	1¼	1⅛	11⅜	10	15⅝
	¾	1	12½	2	25⅜	14½	6¼	1½	1½	11⅜	10	15⅝
	1	1	12½	2	25⅜	14½	6¼	1½	1¾	11⅜	10	15⅝
	1¼	1	12½	2¼	25⅜	11½	6¼	1½	2¼	11⅜	10	15⅝
216	1½	2	15	2¼	31½	16½	7¼	2¼	2⅜	13⅜	12	19⅞
	2	2	15	2¼	31½	16½	7¼	2¼	2⅞	13⅜	12	19⅞
316	2½	2½	19	3¼	36¾	18½	8	3	3	15	12	21
	3	2½	19	3¼	36¾	18½	8	3	3¾	15	12	21

The Webster High-pressure Sylphon Trap

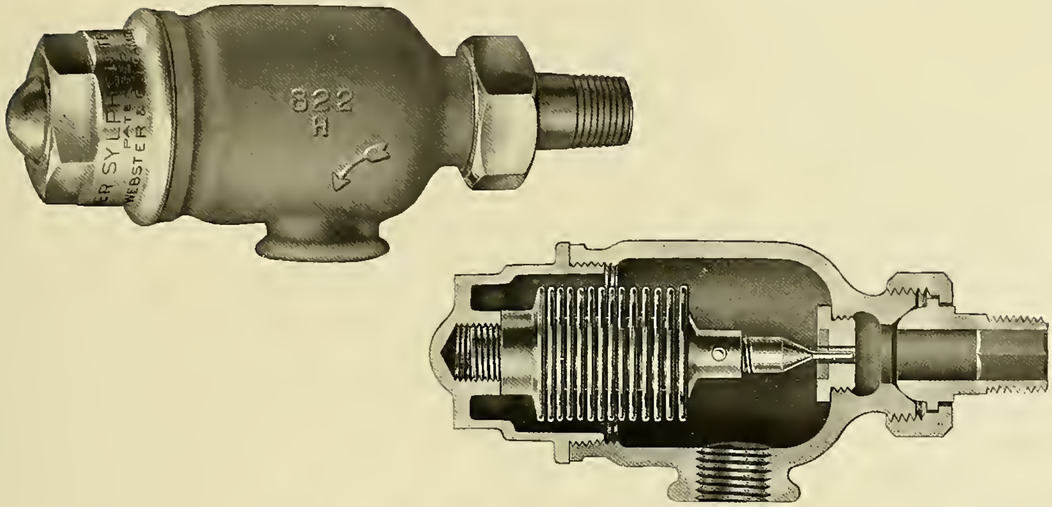


Fig. 24-77. The Webster High-pressure Sylphon Trap

This trap is in many respects like the standard Webster Sylphon Trap described on Page 242. The body construction is the same except that the position of inlet and outlet opening and the union connection of the inlet are reversed.

As the trap must operate at comparatively high steam pressure with resulting high temperature, the thermostatic member or bellows is located outboard of the valve. The sylphon bellows, surrounded in this position with the cooler vapor from the discharged condensate at atmospheric pressure, is extremely sensitive to the much higher temperature of the steam, and consequently acts quickly and positively to close the valve against steam passage through the trap.

It is particularly important when arranging pipe connections that the manufacturer's directions shall be specifically followed.

In consequence also of the higher pressure, the valve piece and the seat are constructed of monel metal, which successfully resists wire-drawing and its accompanying wear.

The Webster High-pressure Sylphon Trap is made in three sizes and for two pressure ranges—Class 2 for pressure from 15 to 50 lb. per sq. in., and Class 3 for pressures to 100 lb. per sq. in.

Application diagrams for this device are shown in Chapters 18 and 20.

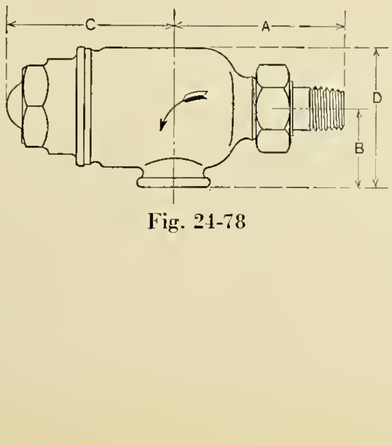


Fig. 24-78

Table 24-22. Dimensions of Webster High-pressure Sylphon Traps

SIZE	A	B	C	D
½″—822	3⅜″	1⅝″	3¼″	2¾″
¾ —833	4 1/16	2⅛	2¾	3⅞
1 —844	4 5/16	2½	3¾	4½

Webster Hydro-pneumatic Tanks

Single and Double-control Types

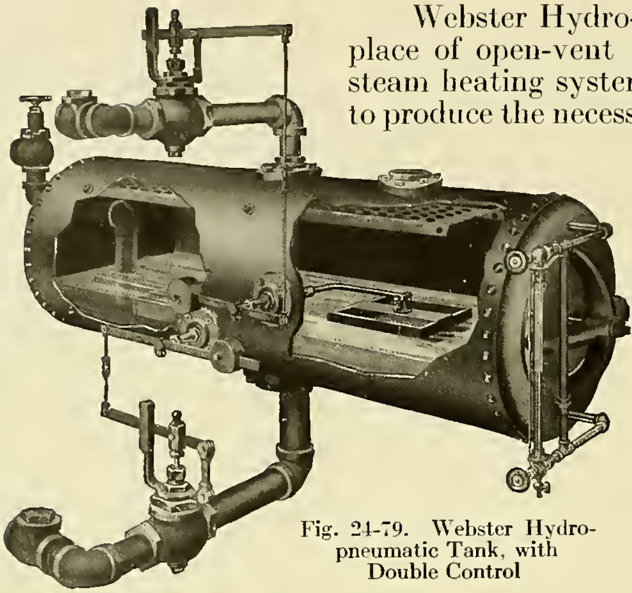


Fig. 24-79. Webster Hydro-pneumatic Tank, with Double Control

Webster Hydro-pneumatic Tanks are used in place of open-vent tanks for receiving returns in steam heating systems where sufficient head room to produce the necessary static head is not available for the installation of a plain receiving tank.

The general design is the same as that of Webster Steam-control and Webster Water-control Receiving Tanks, except that in the Single-control Hydro-pneumatic Tanks the sink pan and rigging control the escape of air through the vent pipe, and in the Double-control type this feature is supplemented by an additional sink pan rigged to control a water valve in the discharge piping.

In both Single and Double-control types the air is permitted to escape freely until the tank is half filled with condensation, when the vent closes and the remaining air is confined. The air vent is open whenever the condensation flows by gravity against the resistance in the outlet connection. When the necessary head is greater than that due to the tank being half full of condensation, the air vent is closed. Further accumulation of air and water creates additional pressure until this, added to the gravity head, overcomes the resistance and condensation flows through the outlet until the water line reaches the middle of tank. Then the air vent opens to permit escape of air. When the tank has no gravity head to heater or boiler the necessary head to overcome resistance in the outlet is by confined pressure only.

The Double-control Hydro-pneumatic Tank, in addition, has its water-control valve arranged to close just before the water level reaches the bottom of the tank. The Double-control type serves to prevent admission of air into the system, through discharge from the tank, when the pressure in the open feed-water heater or boiler may be less than that of the atmosphere.

Both Single and Double-control Tanks are used under pressure greater than the atmosphere and in most instances must be provided with means for preventing excessive pressure due to obstruction of overflow. For this purpose a water-relief valve is provided, which should be piped to an open funnel to facilitate observation and correction of unnecessary waste.

Both Single and Double-control types of tanks are made of riveted flange steel plate, have flat heads and are for installation in horizontal position. A water-trough running along the top distributes the water and assures that sink pans are kept filled with water.

Manholes and covers and gauge glass fittings are regular equipment with both types of tanks. Sizes listed are standard. Others made to order.

Table 24-23. Dimensions of Webster Hydro-pneumatic Tanks

Openings will be bushed to suit requirements. All dimensions in inches

Single-control Type

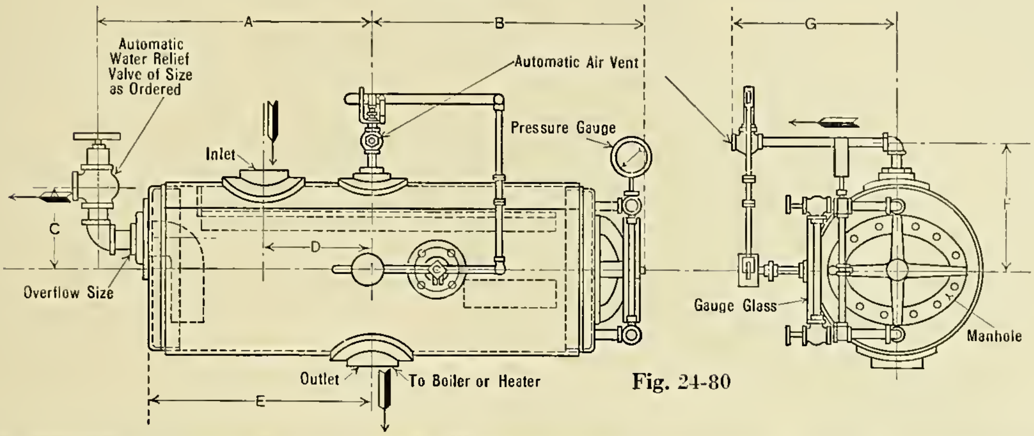


Fig. 24-80

Size	Inlet	Outlet	Vent valve	Overflow	A	B	C	D	E	F	G
18 x 48	4	4	¾	4	29¾	30½	10	12	24¼	14	18
24 x 72	5	5	1¼	5	43¾	42½	13	18	36¼	18	22¾
36 x 96	(2)8	8	1½	6	58	54⅝	18	18	48⅜	24	28¼

Double-control Type

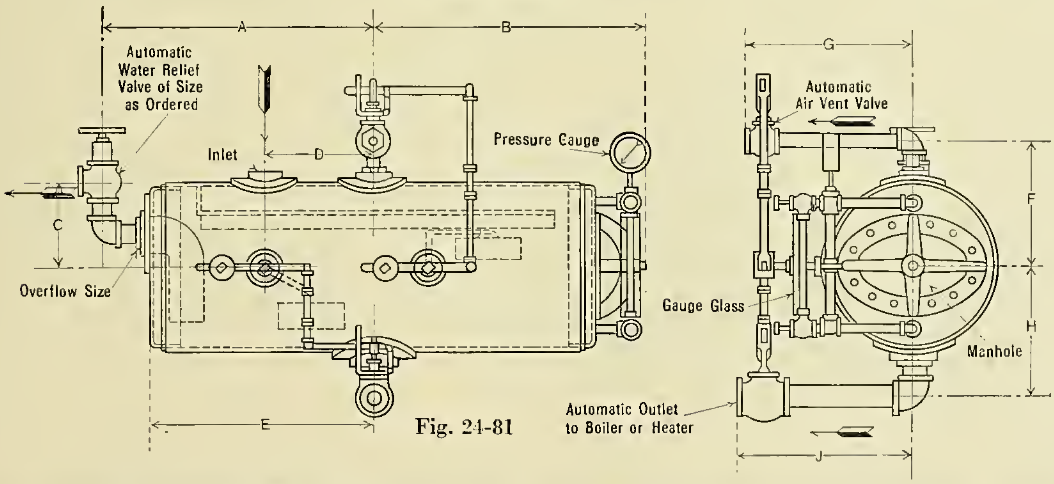


Fig. 24-81

Size	Inlet	Outlet	Vent valve	Overflow	A	B	C	D	E	F	G	H	J
18 x 48	4	4	¾	4	29¾	30½	10	12	24¼	14	18	20	19¼
24 x 72	5	5	1¼	5	43¾	42½	13	18	36¼	18	22¾	22	25¾
36 x 96	(2)8	8	1½	6	58	54⅝	18	18	48⅜	24	28¼	31⅜	35

For ratings, see Table 13-1, page 138.

Webster Expansion Joints

Webster Expansion Joints are constructed with cast-iron bodies and brass-slip sleeves and in both single and double-slip types.

The body of the Webster Expansion Joint is provided with anchors made integral with the body castings for rigid connection to a foundation or a bracket. Service connections are provided for greatest convenience in tapping the steam main for branch piping. Drip outlets also are provided.

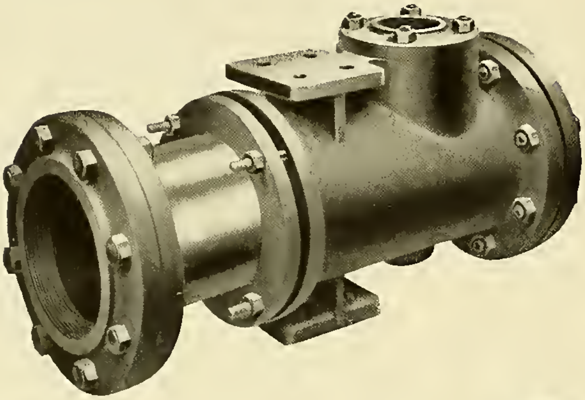


Fig. 24-82. Class D (at left)
Webster Expansion Joint

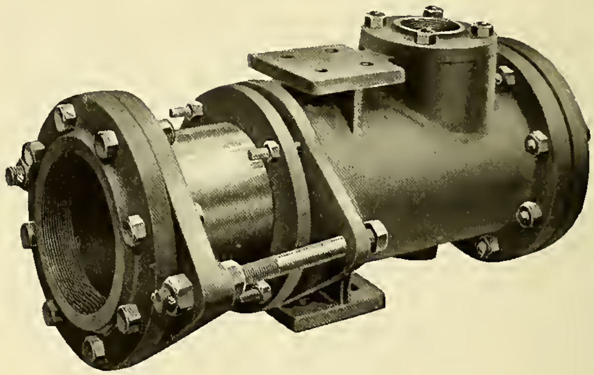


Fig. 24-83. Class DH (at right)
Webster Expansion Joint

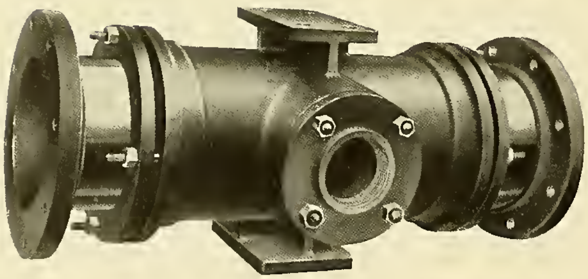


Fig. 24-84. Class G (at left)
Webster Expansion Joint

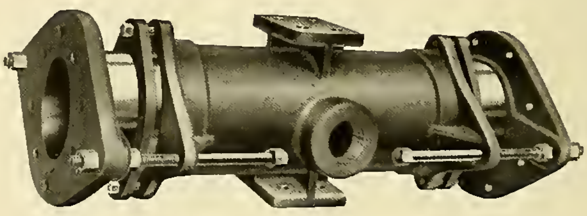


Fig. 21-85. Class GH (at right)
Webster Expansion Joint

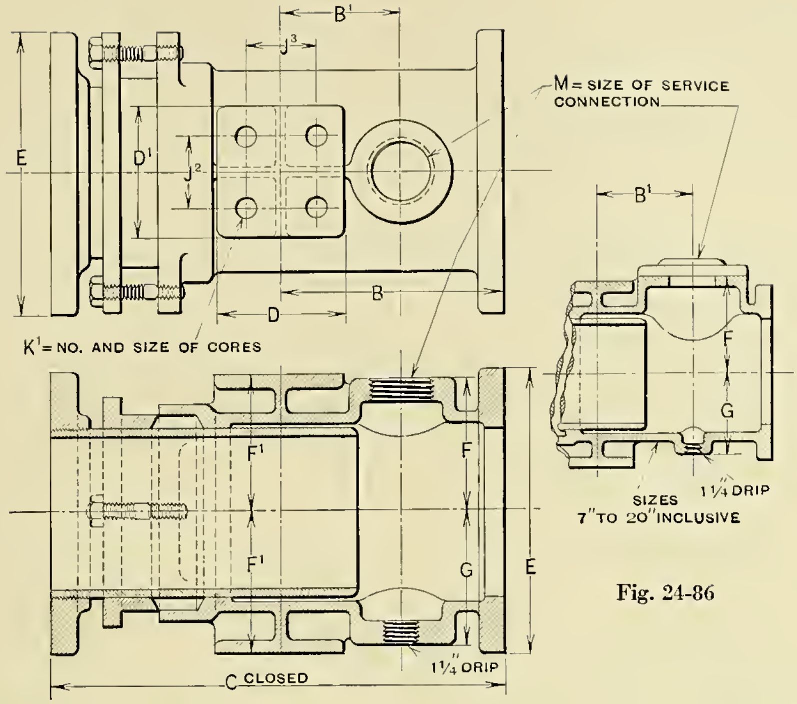


Fig. 24-86

Table 24-24. Class D Webster Expansion Joints for Low-Pressure Steam
Dimensions (in inches)

Size	B	B¹	C	D	D¹	E	F	F¹	G	J²	J³	K¹	M
1½	6¾	4⅜	13⅜	3	3	5	2⅛	2 15/16	2⅛	1¾	1¾	2– ¾	1
2	6	3⅞	13¼	3	3	6	2½	3	2½	1¾	1¾	2– ¾	1¼
2½	6⅝	4⅛	14½	3	3	7	3¼	3½	3¼	1¾	1¾	2– ¾	2
3	6⅝	4⅛	14½	3	3	7½	2⅞	3½	2⅞	1¾	1¾	2– ¾	2
3½	7½	4½	16	4	4	8½	3½	3¾	3½	2	2	4– ⅞	2
4	7½	4½	16	5	5	9	4	4¼	4	2½	2½	4– ⅞	2½
5	8⅜	4⅝	17⅛	5	5	10	4½	5⅛	4½	2½	2½	4– ⅞	2½
6	8¾	4⅞	17¼	6	6	11	5	5⅜	5	3	3	4– ⅞	2½
7	12½	7	20⅞	6	6	12½	6⅝	6½	5¼	3	3	4– ⅞	3
8	13½	7½	21⅞	6	8	13½	7¼	7¼	6	5	3	4– ⅞	3½
10	14½	7¾	23½	6	8	16	8⅝	8½	7	5	3	4–1	4
12	15⅞	8¾	25⅞	7	8	19	9¾	9⅝	8¼	5	3½	4–1	5
14	17½	9¾	28⅛	8	8	21	10¾	10⅝	8¾	5	4	4–1⅛	6
16	17¾	9¾	28½	8	8	23½	12	11⅞	10	5	4	4–1⅛	6
18	18	9¾	28¾	8	12	25	13¼	13⅛	11	9	4	4–1⅛	6
20	18	9¾	30⅝	8	12	27½	14¼	14⅛	12	9	4	4–1¼	6

Maximum working pressure, 15 lb. per sq. in.

This joint has single slip and maximum traverse of 5 in. and is made with a close-grained cast-iron body and brass tubing or cast-brass sleeve.

Standard equipment includes service and drip connections, anchor plates and gland packing.

Companion flanges are furnished only when specially ordered.

Flanges are drilled low-pressure standard unless specially ordered otherwise.

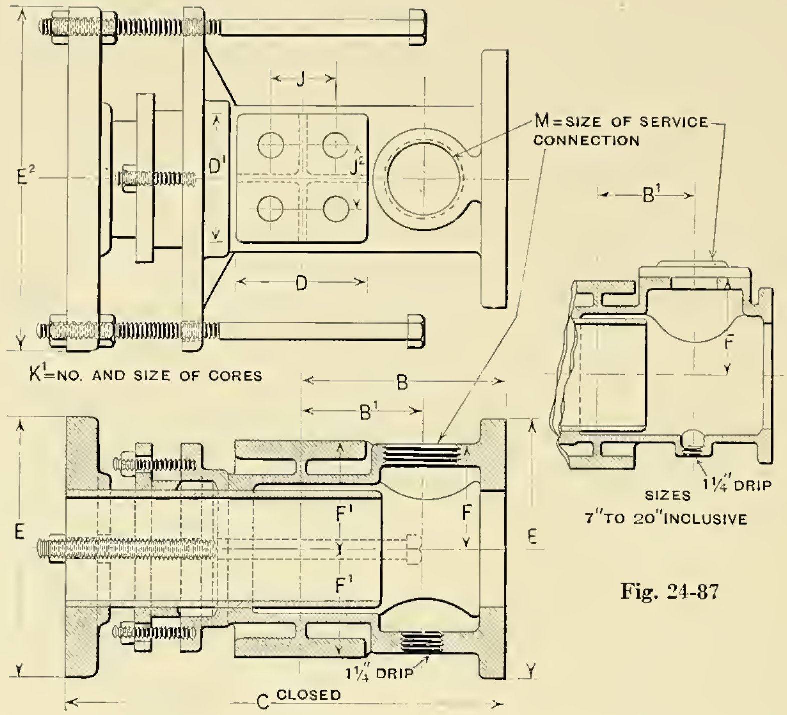


Fig. 24-87

Table 24-25. Class DH Webster Expansion Joints for High-Pressure Steam

Dimensions (in inches)

Size	B	B¹	C	D	D¹	E	E²	F	F¹	J²	J³	K¹	M
1½	6¾	4⅜	13⅜	3	3	5	8¾	2⅛	2 15/16	1¾	1¾	2– ¾	1
2	6	3⅞	13¼	3	3	6	8¾	2½	3	1¾	1¾	2– ¾	1¼
2½	6⅝	4⅛	14½	3	3	7	10½	3¼	3½	1¾	1¾	2– ¾	2
3	6⅝	4⅛	14½	3	3	7½	11	2⅞	3½	1¾	1¾	2– ¾	2
3½	7½	4½	16	4	4	8½	13	3½	3¾	2	2	4– ⅞	2
4	7½	4½	16	5	5	9	12	4	4¼	2½	2½	4– ⅞	2½
5	8⅜	4⅝	17⅛	5	5	10	14¼	4½	5⅛	2½	2½	4– ⅞	2½
6	8¾	4⅞	17¼	6	6	11	15¼	5	5⅜	3	3	4– ⅞	2½
7	12½	7	20⅞	6	6	12½	17¼	6⅝	6½	3	3	4– ⅞	3
8	13½	7½	22⅛	6	8	13½	18½	7¼	7¼	5	3	4– ⅞	3½
10	14½	7¾	23½	6	8	16	21¾	8⅝	8½	5	3	4–1	4
12	15⅞	8¾	25⅞	7	8	19	24¾	9¾	9⅝	5	3½	4–1	5

Maximum working pressure, 125 lb. per sq. in.

This joint has single slip and maximum traverse of 5 in. and is made with a close-grained cast-iron body and brass tubing or cast-brass sleeve.

Standard equipment includes service and drip connections, anchor plates, limit bolts and gland packing.

Companion flanges are furnished only when specially ordered.

Flanges are drilled low-pressure standard unless specially ordered otherwise.

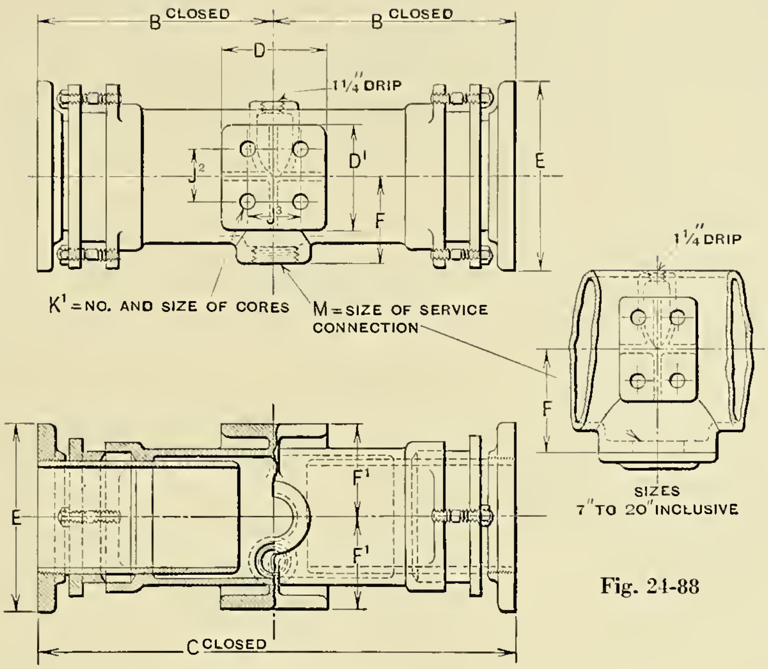


Fig. 24-88

Table 24-26. Class G Webster Expansion Joints for Low-Pressure Steam

Dimensions (in inches)

Size	B	C	D	D¹	E	F	F¹	J²	J³	K¹	M
1½	11 5/16	22⅝	3	3	5	2¼	2 15/16	1¾	1¾	2– ¾	1
2	11⅛	22⅛	3	3	6	2½	3	1¾	1¾	2– ¾	1¼
2½	11⅝	23¼	3	3	7	3¼	3½	1¾	1¾	2– ¾	2
3	12⅝	25¼	3	3	7½	2⅞	3½	1¾	1¾	2– ¾	2
3½	12⅝	25⅛	4	4	8½	3¼	3¾	2	2	4– ⅞	2
4	13⅜	26¾	5	5	9	3½	4¼	2½	2½	4– ⅞	2½
5	13¾	27½	5	5	10	4½	5⅛	2½	2½	4– ⅞	2½
6	13⅞	27¾	6	6	11	5	5⅜	3	3	4– ⅞	2½
7	14⅛	28⅛	6	6	12½	6⅝	6½	3	3	4– ⅞	3
8	15⅞	31¾	6	8	13½	6¾	7¼	5	3	4– ⅞	3½
10	17	33⅞	6	8	16	8	8½	5	3	4–1	4
12	18⅛	36¼	7	8	19	9	9⅝	5	3½	4–1	5
14	18⅞	37¾	8	8	21	10½	10⅝	5	4	4–1⅛	6
16	19½	38⅞	8	8	23½	12	11⅞	5	4	4–1⅛	6
18	20⅛	40⅛	8	12	25	13¼	13⅛	9	4	4–1⅛	6
20	22	44	8	12	27½	14	14⅛	9	4	4–1¼	6

Maximum working pressure, 15 lb. per sq. in.

This joint has double-slip and maximum traverse of 10 in. and is made with a close-grained cast-iron body and brass tubing or cast-brass sleeve.

Standard equipment includes service and drip connections, anchor plates and gland packing.

Companion flanges are furnished only when specially ordered.

Flanges are drilled low-pressure standard unless ordered otherwise.

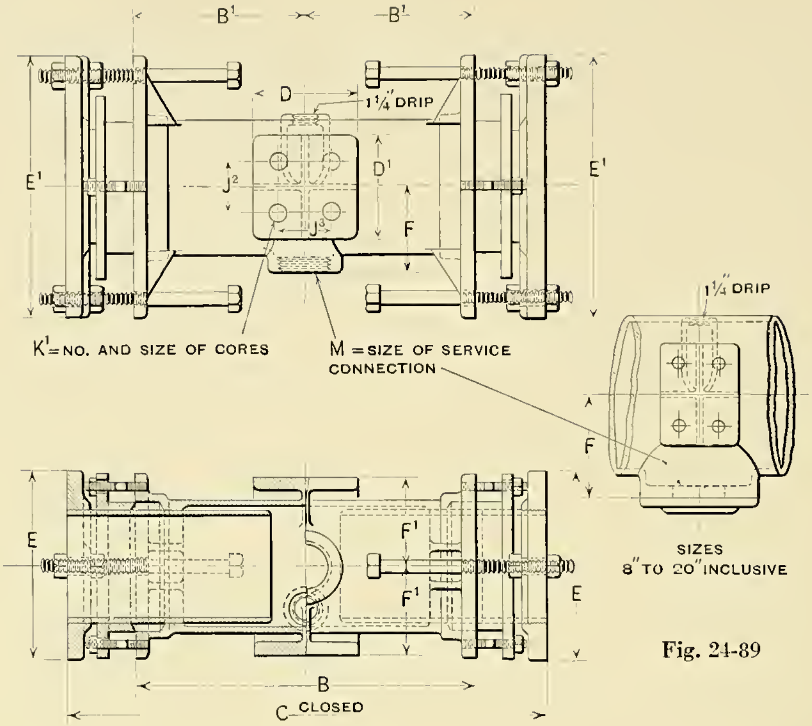


Fig. 24-89

Table 24-27. Class GH Webster Expansion Joints for High-Pressure Steam

Dimensions (in inches)

Size	B	B¹	C	D	D¹	E	E¹	F	F¹	J²	J³	K¹	M
1½	16⅝	8 5/16	22⅝	3	3	5	8¾	2¼	2 15/16	1¾	1¾	2– ¾	1
2	16⅛	8⅛	22⅛	3	3	6	8¾	2½	3	1¾	1¾	2– ¾	1¼
2½	16½	8¼	23¼	3	3	7	10½	3¼	3½	1¾	1¾	2– ¾	2
3	18½	9¼	25¼	3	3	7½	11	2⅞	3½	1¾	1¾	2– ¾	2
3½	18	9	25⅛	4	4	9	13	3¼	3¾	2	2	4– ⅞	2
4	19	9½	26¾	5	5	9	12	3½	4¼	2½	2½	4– ⅞	2½
5	19½	9¾	27½	5	5	10	14¼	4½	5⅛	2½	2½	4– ⅞	2½
6	19¾	9⅞	27¾	6	6	11	15¼	5	5⅜	3	3	4– ⅞	2½
8	23	11½	31¾	6	8	13½	18½	6¾	7¼	5	3	4– ⅞	3½
10	24¼	12⅛	33⅞	6	8	16	21¾	8	8½	5	3	4–1	4
12	25¾	12⅞	36¼	7	8	19	24¾	9	9⅝	5	3½	4–1	5

Maximum working pressure, 125 lb. per sq. in.

This joint has double-slip and maximum traverse of 10 in. and is made with a close-grained cast-iron body and brass tubing or cast-brass sleeve.

Standard equipment includes service and drip connections, anchor plates, limit bolts, and gland packing.

Companion flanges are furnished only when specially ordered.

Flanges are drilled low-pressure standard unless specially ordered otherwise.

Table 24-28. Distance Between Anchor Points and Webster Expansion Joints for Various Steam Pressure Conditions

The following table is recommended as a guide in the design of steam piping for determination of the proper points of installation of Webster Expansion Joints. In such design the *maximum pressure* which the pipe line must sustain during acceptance tests or other special conditions must be selected as the "Gauge pressure"

Gauge pressure	Temperature difference above zero	Expansion Inches per 100 feet	Safe maximum distance in feet between anchors for single-slip expansion joints*
0	212	1.53	260
5.3	227	1.64	245
10.3	240	1.73	225
15.3	250	1.80	220
20.3	259	1.87	215
25.3	267	1.93	210
30.3	274	1.98	202
40.3	286	2.06	195
50.3	297	2.14	190
75.3	320	2.31	175
100.3	337	2.43	166
125.3	352	2.54	160

*For double-slip joints, the safe distance from the joint to an anchor in each direction may be the distance specified for a single joint, provided the body of the double joint itself is securely anchored

Webster Series 21 Steam Separators

For working pressures up to 200 lb. per sq. in.

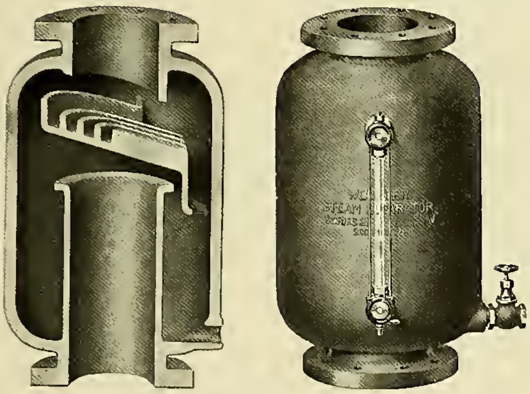


Fig. 24-90. Vertical type

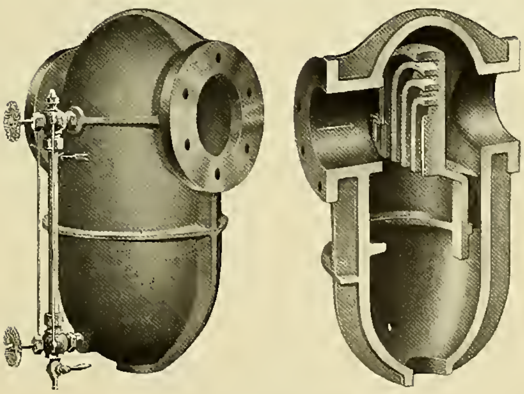


Fig. 24-91. Horizontal type

Webster Steam Separators of the standard types for removing moisture from live steam, have cast-iron corrugated baffles against which the steam impinges, causing a sudden change in direction of flow and consequently freeing the steam of the entrained moisture.

The port openings in every Webster Steam Separator are of such size as to minimize loss of steam pressure from unnecessary friction.

These separators may also be used for special purposes, as removing moisture from compressed air, assuring operation of steam whistles by removing moisture from their steam supplies, etc.

The material ordinarily used in the shells is close-grained cast-iron, but special shells of semi-steel, cast steel or other material can be furnished.

Gauge-glass fittings and drain valves are usual equipment but are furnished only as extras when ordered.

Table 24-29. Dimensions of Webster Series 21 Steam Separators

All dimensions in inches

Companion flanges and gauge and drain fittings furnished only when specially ordered. Flanges are drilled to high-pressure standard unless otherwise ordered

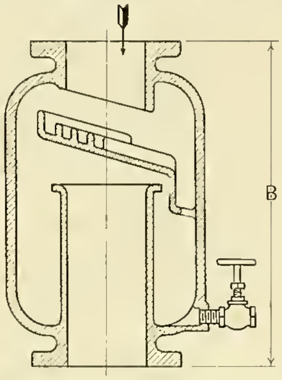
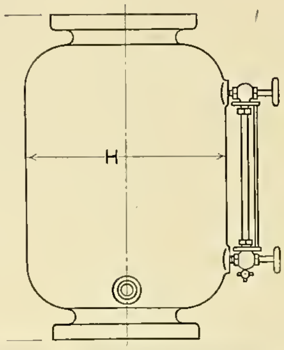


Fig. 24-92. Vertical Type Down-flow only



Dimensions				Flanges		
Size	B	H	Drip	Outside diameter	Bolt circle	No. & sizes of bolts
2	21¼	7	½	6½	5	4– ⅝
2½	22⅜	8	¾	7½	5⅞	4– ¾
3	23½	9½	¾	8¼	6⅝	8– ¾
3½	25⅛	10⅛	1	9	7¼	8– ¾
4	26¾	10¾	1	10	7⅞	8– ¾
5	28½	13½	1	11	9¼	8– ¾
6	30¾	15	1	12½	10⅝	12– ¾
8	33¾	20	1¼	15	13	12– ⅞
10	40¼	23½	1¼	17½	15¼	16–1
12	44¼	27½	1¼	20½	17¾	16–1⅛

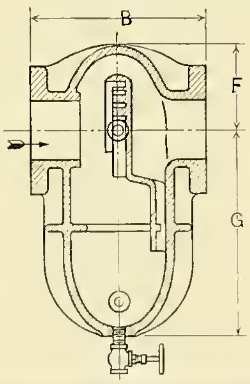
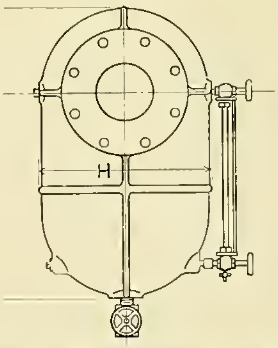


Fig. 24-93. Horizontal Type



Dimensions							Flanges	
Size	B	F	G	H	Drip	Outside diameter	Bolt circle	No. & sizes of bolts
2	9½	3⅞	12½	8⅜	½	6½	5	4– ⅝
3	11½	5¼	14¼	10½	¾	8¼	6⅝	8– ¾
4	13⅛	5¾	16	11½	1	10	7⅞	8– ¾
5	14⅞	7¼	17¾	14½	1	11	9¼	8– ¾
6	16⅝	8⅛	19½	16	1	12½	10⅝	12– ¾
8	20¼	10⅜	23¼	20½	1	15	13	12– ⅞
10	24⅜	12⅜	26¾	24½	1¼	17½	15¼	16–1
12	27½	14¾	30	29¼	1½	20½	17¾	16–1⅛

Special Types of Webster Steam Separators

Working pressures up to 150 lb. per sq. in.

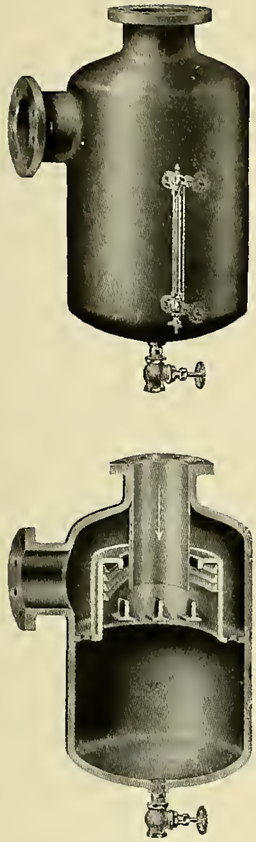


Fig 24-94
Class L—Angle Type
with horizontal outlet

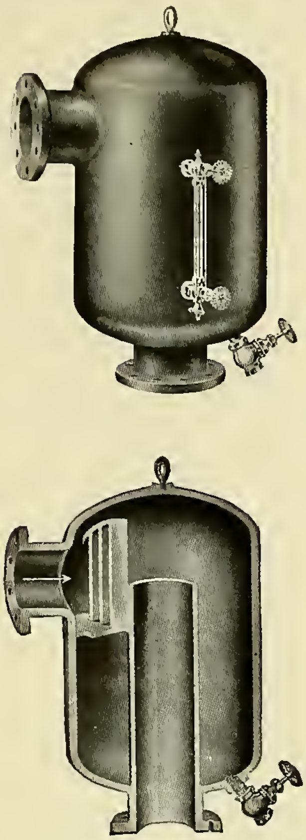


Fig. 24-95
Class M—Angle Type
with bottom outlet

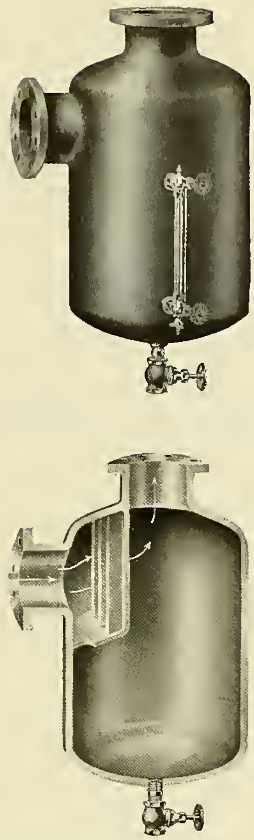


Fig. 24-96
Class N—Angle Type
with top outlet

Table 24-30. Ratings of Webster Steam Separators

Pounds per minute at average gauge pressures. Based upon a pipe velocity of 6000 ft. per min.

Size of separator in.	Gauge pressures 100 Lb. per sq. inch	125 Lb. per sq. inch	150 Lb. per sq. inch	200 Lb. per sq. inch
2	35.	43.3	51.6	66.6
3	78.3	96.7	112.	141.
4	140.	167.	196.	250.
5	215.	258.	300.	391.
6	317.	383.	450.	583.
7	433.	516.	600.	783.
8	550.	660.	800.	1000.
10	883.	1083.	1250.	1580.
12	1250.	1533.	1800.	2333.

CHAPTER XXV

Specifications for Webster Systems

THE following specifications cover typical Webster Systems only in a general way, and are subject to many variations. It is advised that wherever practical a Webster Service Engineer be called into consultation during the preparation of plans and specifications for Webster Systems.

Specifications for the Webster Vacuum System of Steam Heating

(This specification is written for a system of the usual up-feed type. For the variations known as the Webster Conserving System and the Webster Hylo System, revised special clauses will be furnished by Warren Webster & Company on application.)

GENERAL: (Here specify the general requirements of the contract, such as intent of drawings and specifications; verification of measurements; co-operation with other contractors; foreman; ordinances; permits; protection of work and buildings; rights reserved; extra work; return of specifications and drawings; payments, etc.)

CUTTING OF FLOORS AND WALLS: The [building] [heating] contractor will cut all holes in floors and walls and provide trenches and covers for piping which may be necessary for this work, and at completion make all repairs to floors and walls so cut.

SCOPE OF WORK: This specification is intended to cover a 2-pipe low-pressure heating system known as the Webster Vacuum System of Steam Heating.

It is intended to supply radiation for heating the building to a temperature of .... degrees fahr. when the outside temperature is.....or a corresponding equivalent difference in temperature, with doors and windows reasonably tight.

SPECIAL APPARATUS: The basis of this specification being a Webster System, each bidder is required to submit his proposal for furnishing apparatus manufactured by Warren Webster and Company.

*ALTERNATE PROPOSALS will be considered for the use of modulation supply valves and thermostatic return traps of the same supply and return tappings and as made by....... ...., provided each bidder states in his proposal the sum which will be added to or deducted from his main bid in case they are used.

STANDARD APPARATUS: In addition to the special apparatus, this contractor is to furnish all other material and labor necessary for the complete work as shown on plans or called for in specifications.

RADIATION: All pipe coils must be made up of full-weight [mild-steel] [genuine wrought iron] pipe and best gray-cast iron fittings and manifolds. All radiators must be of the .... pattern equal in every respect to that manufactured by .... and must be of the heights and columns shown on plans. They must be of the [steam] [hot-water] type.

(*Note:* If of hot-water pattern, specify that the radiators "shall be connected with the supply at the top and the return at the diagonally opposite lower corner." If of steam type, specify that they "shall be provided with eccentric bushings and connected so that the bottom of the return connection will be lower than the bottom of the supply connection." Where Webster Modulation Valves are to be used, hot-water type radiators should be specified.)

* To be inserted in case the Architect or Engineer desires to obtain, for comparative purposes, an alternate price upon apparatus of a make other than Webster

Contractors supplying radiation ordered for this work shall, if they be called upon to do so, demonstrate to the satisfaction of the owners or their authorized representative, that the radiation furnished contains in each section of the different types supplied the amount of prime heating surface mentioned in the lists published by the manufacturers of the respective types. This must be demonstrated by actual measurement and the development of the exposed surface of the sections.

The heating contractor is to instruct the manufacturer of the radiation that he requires same to be thoroughly pickled and cleaned before shipment and that the outlets are to be plugged with loose wooden plugs. The manufacturer must issue his certificate to the contractor showing that these radiators have been so cleaned. These radiators are to be kept plugged until same are connected to the different pipe lines.

Air-valve tappings are to be plugged.

Radiators must be tapped or bushed for sizes of supplies and returns as shown on plans.

Coil Hangers: Overhead radiators are to be hung in special pipe hangers and in no case shall these coil hangers be more than 10 ft. apart.

Wall coils are to have spring pieces and are to be hung on cast or wrought-iron plates spaced as directed by their manufacturer, screwed to 1½-in. strap-iron brackets bent to shape, and securely fastened to the walls with two expansion bolts each. Brackets must be spaced not over 10-ft. centers. Wall radiators must be hung as directed by manufacturers.

Straps shall be painted two coats of lead and oil paint of colors as directed by owners before radiators are set in place. Owners must be given opportunity to paint walls or ceilings before radiators are set.

Return Traps: The return end of every radiator, pipe coil or other form of heating unit must be provided with a Webster Return Trap (of the type selected). The size of the trap shall be governed by the amount of condensation from the radiation unit as called for on plans. The connections of Webster Return Traps must be made to the approval of Warren Webster & Company, who will provide the contractor with service details showing approved forms of connection.

Supply Valves: Each radiation unit must be provided with a Webster Modulation Valve connected to the top supply tapping.

The sizes of supply valves, the radiator tappings and the sizes of horizontal branches from risers to radiators must be as shown on the plans, or as hereafter described in this specification.

Pipe: All low-pressure pipe must be full-weight [mild-steel] [genuine wrought-iron] equal to that manufactured by .... All screwed piping must be fitted with occasional flanged unions. Where supply pipes are reduced in the run, eccentric reducing couplings must be used.

Straighten all pipe, ream all burrs and remove all dirt before erecting pipe or fittings. Have all runs plumb and parallel with building. Provide Webster Expansion Joints of the types and sizes and at the points shown on plans. Support all pipes securely and in such manner as to permit unobstructed movement between anchorages for expansion and contraction.

So far as possible, all horizontal runs must be graded in the direction of steam flow.

Fittings: All fittings shall be best gray-iron, straight and true and free from blowholes or other defects; equal to those manufactured by .... Fittings for low pressure shall be standard weight; those for high pressure shall be extra heavy.

Valves: All check, gate and globe valves must be equal to those manufactured by ....

Heat Mains: From the low-pressure side of pressure-reducing valve run a pipe to connect into the exhaust steam main where shown on plans. (Here should follow a description of the course of the steam main and its branches.)

Horizontal runs must grade not less than 1 in. in 25 ft.

Live Steam Connection: Connect a ...-in. line from outlet in live steam main (where indicated on plans) to the heating main through the pressure-regulating valve. This valve shall be ...-in. size and equal to that manufactured by ...., and shall be set to reduce the steam pressure from ... to (1 lb. per sq. in. or less).

Provide a 3-valve bypass as shown, the valve in front of the reducing valve to be

of the globe pattern. Run a "control pipe" as shown. Place a low-pressure gauge and a $\frac{3}{4}$-in. pop alarm valve set at 10-lb. pressure in the heat main about 10 ft. from the discharge of pressure-reducing valve.

RISERS: A system of supply and return risers is to be run as shown on plans. Risers are to be run [exposed] [concealed] and are to be of sizes marked on plans. All radiator branches must grade back to risers or mains with as much grade as possible, in no case less than 1 in. in 5 ft. All connections are to be made with ample provision for expansion and contraction *and particular care is to be taken that branches are run without pockets.*

RETURN PIPING: All return risers and branches are to connect into return mains. Horizontal return piping must be graded toward the vacuum pump not less than 1 in. in 40 ft.

DIRT TRAPS: The bottom of all supply connections taken from the heating main must be dripped into the vacuum return by means of a cooling leg, a gate valve, a Webster Dirt Strainer and a Webster Return Trap of size shown on plans.

Note: In large installations it is advisable to run a separate gravity drip line and connect drip of each riser or drip point of main through $\frac{3}{4}$-in. line with gate valve to this line. The discharge of this gravity drip line to be to the feed-water heater through loop seal or to the vacuum return through Webster Heavy-duty Trap.

DIRT STRAINERS: Provide and connect Webster Dirt Strainers of the sizes specified and at the points indicated on the plans.

LIFT FITTINGS: Where lifts occur in the vacuum return lines they are each to be provided with a pair of Webster Lift Fittings of the sizes called for on plans and connected according to special service detail furnished by the manufacturer.

BOILERS: (Here specify the make, size and type of boiler or boilers required; also the equipment required for the complete boiler plant, including smoke breeching, damper regulator, gauges, feed pump, injector and any other necessary accessories.)

VACUUM PUMPS: (Here specify the make, size, type and number of pumps required "to be furnished upon (concrete or other material) foundations to be provided by this contractor." Detail specifications of pumps should describe either the electric-driven type (Nash, etc.) or the steam-driven type (Blake-Knowles, Burnham, Marsh, etc.). For steam-driven pump, specify "simplex, double-acting type, brass lining, and fitted for hot-water service" and that "each pump shall be provided with a forced-feed lubricator of approved make and having a capacity of one quart."

Each pump shall have ample capacity for handling the products of condensation from the entire heating system.

The discharge from steam-driven vacuum pump must be connected to the proper tapping in the receiving tank. If discharge outlet is located on the side of the steam pump, tap the cover plate above the discharge valves and run $\frac{3}{4}$-in. air line, connecting to discharge pipe.

All connections must be properly valved and made complete.

SUCTION STRAINER: In the suction pipe to the vacuum pump, place a Webster Suction Strainer. This strainer must be connected to accord with special service detail furnished by the manufacturer.

VACUUM GOVERNOR: In the steam connection to vacuum pump below the lubricator there must be placed a . . .-in. Webster Vacuum-pump Governor with 3-valve bypass. Same must be connected by means of $\frac{1}{2}$-in. vacuum line to the suction strainer and also to the vacuum gauge on board. Each branch must be provided with a globe valve.

GAUGES: Furnish and erect at convenient position two $5\frac{1}{2}$-in. compound gauges mounted on a slate board. Connect one gauge to equalizing line between heat main and reducing valve, one gauge to a line connecting vacuum governor with vacuum return at suction strainer. All gauge piping to be $\frac{1}{2}$-in. and all branches valved.

AIR-SEPARATING TANK: Furnish a Webster Air-separating Tank . . . in. in diameter by . . . in. long. This tank is to be of the. . . . . .type.

Erect the separating tank as high above the heater as possible, as shown on plans, and to it make connections from discharge of vacuum pumps and to feed-water heater through long loop seal.

From top outlet on tank make a vent connection to atmosphere.

Feed-water Heater: Furnish and erect on foundation one Webster Feed-water Heater of sufficient capacity for heating the required feed water to within 5 deg. of the temperature of the steam entering same.

The drip from oil separator is to connect to waste line through a Webster Grease Trap with 3-valve bypass and check valve as shown in special service detail.

The contractor is to make all necessary steam, water and drain connections as shown or called for.

Steam Separator: Furnish and connect Webster Steam Separators of approved type to steam lines as shown or called for.

The drip from bottom of each separator is to be connected into a high-pressure trap of approved make. Each trap is to be provided with a 3-valve bypass. The discharge lines from these traps are to be connected into the feed-water heater.

Covering: After all piping and apparatus have been tested and made tight to the approval of the [architect] [engineer] or his representative, the following covering is to be applied. (Here specify necessary covering for boilers, heater, separator, and all [specify which] piping, valves and fittings.)

Painting and Bronzing: All radiators, coils and exposed piping throughout the building, after being tested, are to be painted or bronzed as follows:

All radiators, coil and exposed piping are to be painted one coat of sizing and then [bronzed] [painted] [two] coats; color as selected by architect or owner.

All exposed parts of boiler and heater to be painted two coats of black asphaltum paint.

Tests: All concealed pipes and risers shall be tested and made tight under an hydraulic pressure of 50 lb. per sq. in. before being covered in. The entire system shall be tested and made tight under 10-lb. steam pressure.

Thoroughly blow out the pipes to free them from all accumulation of dirt, chips and other material, making temporary piping connections for this purpose.

Fuel and Labor: The heating contractor will furnish all fuel and labor required for testing and adjusting boilers and apparatus and for drying out covering on boilers (and smoke breeching). He will also remove water and ashes resulting therefrom.

Temporary Setting of Radiators: Upon written request of the [architect] [engineer] the contractor shall connect up for temporary heat such radiators as shall be designated. These radiators shall afterwards be disconnected, moved, cleaned, and afterwards reconnected permanently. Wall radiators and radiators without leg sections shall be supported on wooden blocks. Each radiator is to have two pipe connections and no supply or return valves are to be attached at this time. Each bidder will state in his proposal a unit price which he will charge for making temporary connections as described above.

Inspection: This job is to be inspected by a representative of the manufacturer of the return traps before acceptance and he shall submit a written report of the same to the Architects.

Guarantee: The contractor must agree to make good at his own expense any defects in labor or material furnished by him for this work which may develop within one year from the completion of this contract, reasonable wear and tear excepted.

The entire system when completed is to be tested in the presence of the [architect] [engineer] or his representative, and made tight without caulking. The contractor will be held liable for any damage to the building or its contents due to leaks or other defects in his work which may develop during the period of installation and test.

Specifications for the Webster Modulation System of Steam Heating

(This specification is written for a large residence. It is, of course, subject to modifications and variations for other kinds of buildings, for other sources of steam than house boiler, etc., for which revised typical specification clauses will be furnished by Warren Webster & Company on request.)

General: (Here specify the general requirements of the contract such as intent of drawings and specifications; verification of measurements; co-operation with other con-

tractors; foreman; ordinances; permits; protection of work and buildings; rights reserved; extra work; return of specifications and drawings; payments, etc.)

Cutting of Floors and Walls: The [building] [heating] contractor will cut all holes in floors and walls and provide trenches and covers for piping which may be necessary for this work, and at completion make all repairs to floors and walls so cut.

Scope of Work: This specification is intended to cover a 2-pipe open-return heating system known as the Webster Modulation System of Steam Heating.

It is intended that sufficient radiation shall be supplied for heating the building to a temperature of ... deg. fahr. when the outside temperature is ... deg. fahr. or a corresponding equivalent difference in temperature, based upon all doors and windows being fitted reasonably tight to prevent excessive infiltration of cold air.

Special Apparatus: The basis of this specification being a Webster System, each bidder is required to submit his proposal for furnishing apparatus manufactured by Warren Webster and Company.

*Alternate Proposals will be considered for the use of modulation supply valves and thermostatic return traps of the same supply and return tappings and as made by ...., provided each bidder states in his proposal the sum which will be added to or deducted from his main bid in case they are used.

Standard Apparatus: In addition to the special apparatus, this contractor is to furnish all other material and labor necessary for the complete work as shown on plans or called for in specifications.

Boilers: (Here specify the make, size and type of boiler or boilers required, specifying also the equipment required for the complete boiler plants, including smoke breeching and other necessary accessories.) (Indicate what contractor is to build boiler foundation.)

Note: Boilers and auxiliary equipment must be installed in accordance with Warren Webster & Company's standard service details.

Damper Regulator: Furnish one Webster Damper Regulator for each boiler; to be connected in accordance with the manufacturer's standard details.

Gauges: A special compound gauge for Webster Modulation System is to be installed for each boiler. This gauge will be furnished by the manufacturers of the system.

Radiators: All radiators throughout the building shall be of .... or equal approved make; all radiators to be of the hot-water type with supply tapping at top and return tapping eccentric at diagonally opposite lower corner. Radiators to be of the height and columns and to contain the surface indicated on plans. In no case is radiation to project above window sill. In connecting all radiators, the inlet end shall be placed next to feed risers, if possible.

The indirect stacks are to be .... (make and type) cast-iron radiation, to be of the size and contain the number of sections as called for on plans.

The heating contractor is to instruct the manufacturer of the radiation that same is to be thoroughly pickled and cleaned before shipment and that the outlets are to be plugged with loose wooden plugs. The manufacturer must issue his certificate to the contractor showing that these radiators have been so cleaned. These radiators are to be kept plugged until they are installed and connected.

Air valve tappings are to be plugged.

Radiators must be tapped or bushed for sizes of supplies and returns as shown on plans.

Hangers: Hangers for indirect stacks are to be strong wrought-iron or pipe supports.

Enclosures for Radiators: The enclosures and grilles for enclosed radiators will be furnished by ....

Return Traps: The return end of every radiator, pipe-coil or other form of heating unit must be provided with a Webster Return Trap (of the type selected). The size of the trap for each radiation unit shall be as shown on plan or called for in specification. The connections of Webster Return Traps must be made to the approval of Warren Webster & Company, who will provide the contractor with service details showing approved forms of connection.

Supply Valves: Each radiation unit must be provided with a Webster Modulation Valve connected to the top supply tapping.

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* To be inserted in case the Architect or Engineer desires to obtain, for comparative purposes, an alternate price upon apparatus of a make other than Webster

The sizes of supply valves, the radiator tappings and the sizes of horizontal branches from risers to radiators must be as shown on plans.

Each overhead radiator must be provided with a Webster Modulation Valve with chain attachment.

Provide a Webster Modulation Extended-stem Valve for each radiator behind a grille.

Modulation Vent Trap: Furnish and install [one] No..... Webster Modulation Vent Trap for separating the air from the condensation in the heating system. [The] [Each] trap is to be vented through ...... Webster Vent Valve[s] placed in the top.

Pipe: All pipe must be full-weight [mild-steel] [genuine wrought iron] equal to that manufactured by .... All screwed piping must be fitted with occasional flanged unions. Where supply pipes are reduced in the run, eccentric reducing couplings must be used.

Straighten all pipe, ream all burrs and remove all dirt before erecting pipe or fittings. Have all runs plumb and parallel with building. Allowance for expansion and contraction must be provided. Support all pipes securely and in such manner as to permit unobstructed movement between anchorages for expansion and contraction.

So far as possible, all horizontal runs must be graded in the direction of steam flow; where this is not possible, the pipe lines shall be materially increased in size as shown on plans.

Fittings: All fittings shall be best gray-iron, straight and true and free from holes or other defects; equal to those manufactured by .... Fittings shall be standard-weight.

Valves: All gate valves must be equal to those manufactured by .... All check valves must be special, of balanced type with vertical seat, and of approved make.

Fresh-air Inlets: Fresh-air inlets for indirect heating are to be taken from openings provided in walls. Another contractor will provide heavy copper wire screens having ½-in. mesh, and sheet metal louvers over the mouth of each inlet.

Sheet Metal Work: The ducts supplying fresh air to the indirect stacks, the indirect stack casings and the hot-air flue from indirect stacks to registers are to be made of galvanized iron. They are to be properly braced and locked tight to prevent air leakage. An adjustable lock quadrant hand damper is to be provided in cold-air connection to each indirect stack.

The metal used for all ducts and flues is to conform to the following gauges:

Ducts that have one dimension over 48 in., ... gauge.

Ducts that have one dimension from 30 to 48 in., ... gauge.

Ducts that have one dimension from 12 to 30 in., ... gauge.

Ducts that have one dimension smaller than 12 in., ... gauge.

The indirect stack casings are to be made of ... gauge iron and are to be built neatly around stacks and provided with cleanout doors above and below radiators in bottom or side.

Registers: The registers for the outlets of hot-air flues from indirect stacks will be furnished by ....; their installation is included within this contract.

Steam Piping: From the steam outlets on boiler rise and connect to a steam header over boiler. From top of header take branches as shown. The steam lines are to be run close to ceiling of cellar with a grade of 1 in. in 25 ft. The branches for risers are to be taken from top of mains. Steam header and main are to be dripped to wet drip line where shown.

Risers: A system of supply and return risers is to be run as shown. Risers are to be run [exposed] [concealed], and are to be of sizes marked on plans. Unless otherwise noted on plans, branches to radiators above first floor are to be run concealed in floor construction and branches to first floor radiators are to be run overhead in cellar as close to ceiling as possible. All radiator branches are to grade back to risers or mains with as much grade as possible, in no case less than 1 in. in 5 ft. All connections are to be made with ample provision for expansion and contraction and *particular care is to be taken that branches are run without pockets.*

Return Piping: All return risers and returns from first floor radiators are to connect into overhead return mains. The return mains are to start as high as possible and grade toward the Webster Modulation Vent Trap 1 in. in 25 ft. *The vent trap (or traps) to be located where shown and at least 30-in. above the water line and as much higher as possible.*

[The] [Each] vent trap will be provided with a tapping near the top into which the dry return main must be connected, a tapping in the bottom from which a ... in. pipe must be run to below boiler water line and connected into the wet return through a horizontal swing check valve of .... make. Make a full size bypass connection around [each] vent trap. Make a ...-in. city water supply connection to boiler with check valve and cock, also a ...-in. drain to waste through gate valve from the return header of boiler as directed. Check valves are to be installed where shown.

A wet drip line is to be run on wall near floor as shown, and connected to boiler. To this line connect drips of mains, indirect radiators and lines from vent trap as shown.

Covering: After all piping and apparatus has been tested and made tight to the approval of the [architect] [engineer], the following covering is to be applied. (Here specify necessary covering for boilers, and all steam, return and drip piping, valves and fittings.)

Painting and Bronzing: All radiators, coils and exposed piping throughout the building, after being tested, are to be [painted] [bronzed] as follows: All radiators, coils and exposed piping throughout the building are to be painted one coat of sizing and then bronzed or painted [two] coats; color as selected by architect or owner.

All exposed parts of boiler to be painted two coats of black asphaltum paint.

Radiators or ducts which are visible through grilles or registers are to be painted two coats of dull black.

Tests: All concealed pipes and risers shall be tested and made tight under an hydraulic pressure of 50 lb. per sq. in. before being covered in. The entire system shall be tested under 10 lb. steam pressure. The entire system shall be thoroughly washed out before final test, wasting condensation to sewer or other point of disposal.

Cleaning Boilers: Remove safety valve, place inside the boiler a sufficient quantity of soda ash to cause saponification of oils and grease. Run temporary overflow pipe to waste, from safety valve outlet or from highest point of boiler and start moderate fire so that foaming of boiler will cause flow of oil and grease to waste, at the same time feeding the boiler with water to prevent injury to same. After thoroughly boiling out the boiler, draw the fire and when cool draw off all water from the boiler and thoroughly wash same with clean water to remove dirt and chemicals. The treatment of boiler should be repeated if water line fluctuates abnormally or shows signs of foaming.

Fuel and Labor: The heating contractor will furnish all fuel and labor required for testing and adjusting boilers and apparatus and for drying out covering on boilers (and smoke breeching). He will also remove water and ashes resulting therefrom.

Temporary Setting of Radiators: Upon written request of the [architect] [engineer] the contractor shall connect up for temporary heat such radiators as shall be designated. These radiators shall afterwards be disconnected, moved, cleaned, and afterwards reconnected permanently. Wall radiators and radiators without leg sections shall be supported on wooden blocks. Each radiator is to have two pipe connections and no supply or return valves are to be attached at this time. Each bidder will state in his proposal a unit price which he will charge for making temporary connections as described above.

Inspection: This work is to be inspected by a representative of the manufacturer of the return traps before acceptance and he shall submit a written report of the same to the Architects.

Guarantee: The contractor must agree to make good at his own expense any defects in labor or material furnished by him for this work which may develop within one year from the completion of this contract, reasonable wear and tear excepted.

The entire system when completed is to be tested in the presence of the architect or his representative, and made tight without caulking. The contractor will be held liable for any damage to the building or its contents due to leaks or other defects in his work which may develop during the period of installation and test.

CHAPTER XXVI

Webster Sylphon Trap Attachments

1. For "Sylphonizing" Webster Traps of Earlier Types

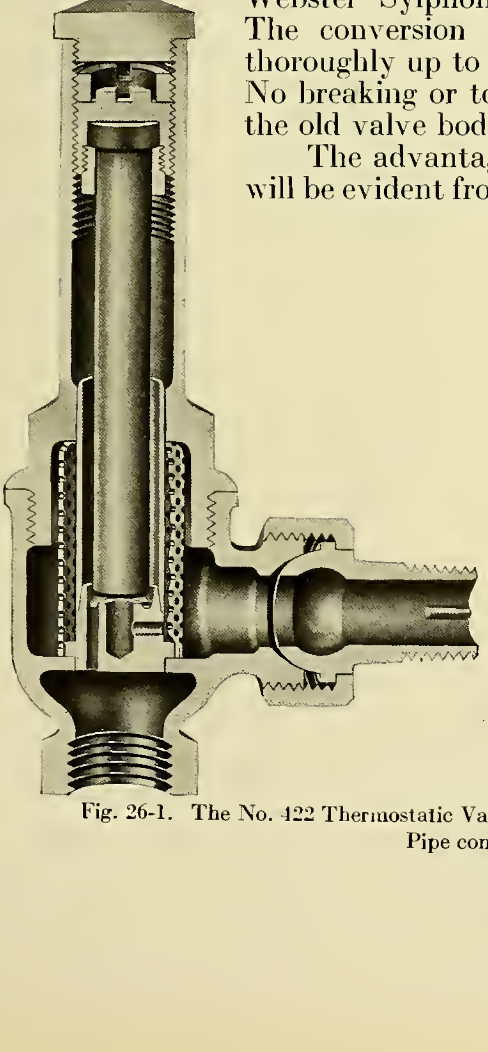
STEAM heating, like almost every other science, has developed progressively through experience.

Being pioneers in this field Warren Webster & Co. have had ample incentive and opportunity for experimental research and development, and have constantly improved their product and methods, discarding and abandoning earlier types of apparatus as improved forms were adopted.

The Webster Sylphon Trap (shown and described on pages 242-5) is now generally recognized by leading architects and engineers to be the most satisfactory type of device for return line systems. It is in its eleventh year of success and the total number in use has passed the million mark.

Owners of buildings and plants in which old-style Webster Valves are in use will be vitally interested in knowing that such valves can be readily converted into Webster Sylphon Traps by means of the Webster Sylphon Attachments described in this chapter. The conversion necessary to bring the heating system thoroughly up to date can be made at a very moderate cost. No breaking or touching of pipe connections is involved, as the old valve bodies are utilized.

The advantages to be derived from the "change over" will be evident from the description of the Webster Sylphon



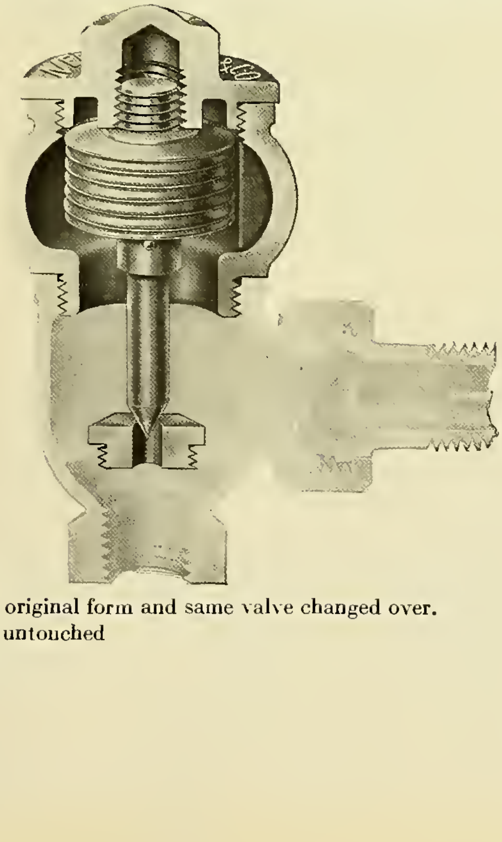


Fig. 26-1. The No. 422 Thermostatic Valve in its original form and same valve changed over. Pipe connections untouched

Trap, on page 242, which description will equally fit the earlier Webster Valves after they are converted by means of Webster Sylphon Attachments. The time required for changing over any valve is only a few minutes.

Conversion of No. 422 Webster Thermostatic Valves: The method of changing over by means of the 5-A-13 Webster Sylphon Attachment is indicated by the illustrations.

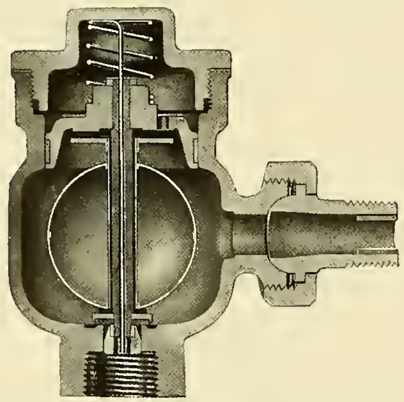
It is only necessary to remove the old bonnets and interior parts, tapping the body for the insertion of a new brass seat by means of a tapping tool. The Webster Sylphon Trap Attachment may then be inserted and the old valve has become a new Webster Sylphon Trap equal in performance to the standard Webster Sylphon Traps which are furnished to thousands of new customers each year.

For conversion of Multiple-unit Thermostatic Valves, see page 296.

Conversion of Webster Motor Valves: This is practically the same as with the No. 422 Webster Thermostatic Valve except that a slightly different Sylphon Attachment is used.

The illustrations show the No. 522 M Sylphon Attachments for ½-in. motor-valves of the disc-air-port type. The No. 533 M Attachment for ¾-in. motor-valves is of exactly the same construction. These same Sylphon Attachments may be applied to the '03 motor-valves of the pin-air-port type where this special type of valve is to be changed over.

For conversion of Multiple-unit Motor Valves, see page 296.



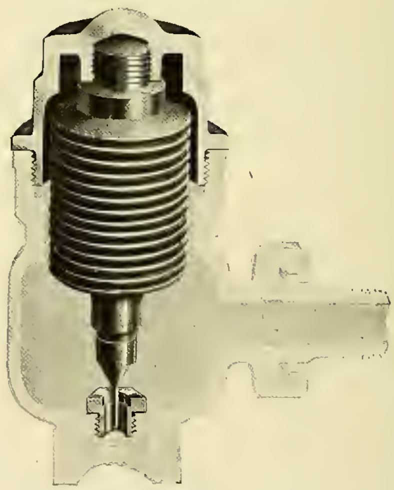


Fig. 26–2. ½-Inch Webster Motor-Valve, Disc-Port Type, in its original form and same valve changed over. Pipe connections untouched

Conversion of No. 422 Webster Water-seal Motors: The method of changing over, as illustrated, involves the use of the same attachment as for changing over the Webster Thermostatic Valve as just described. In the case of the Water-seal Motor, however, the operation is simplified through the old body being already tapped for the valve seat.

It is only necessary to remove the old bonnets and interior parts, and insert the new brass seat. The Webster Sylphon Trap Attachment may then be inserted and the old valve becomes a new Webster Sylphon Trap.

For conversion of Multiple-unit Valves, see page 296.

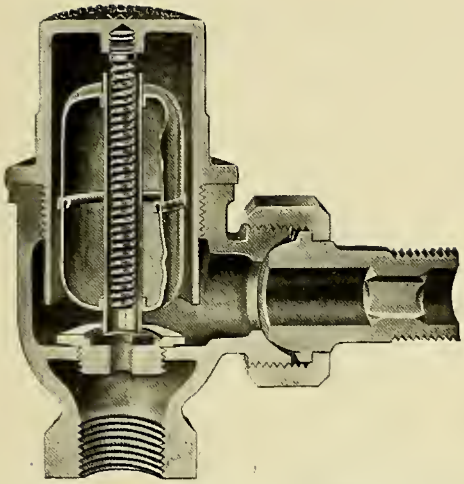
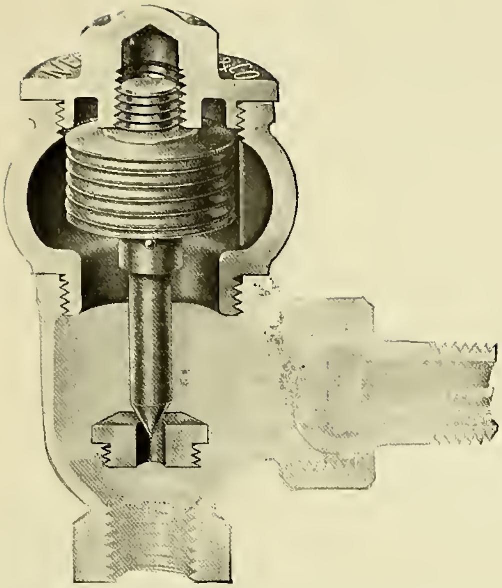


Fig. 26-3. The No. 422 Water-seal Motor in its original form and same motor changed over. Pipe connections untouched



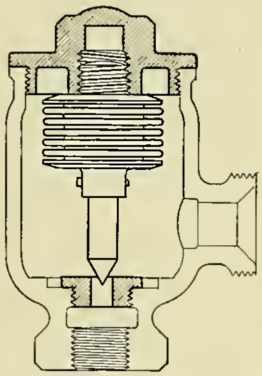


Fig. 26-4. No. 5-C-15 Sylphon Attachment for 522 or 523 Water-seal Trap where the discharge rating is low

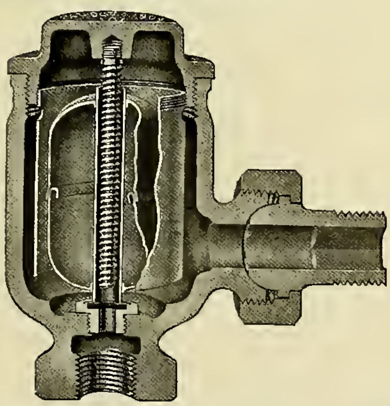
Conversion of No. 522 Water-seal Traps: The change-over in this instance requires only removal of the old bonnets and interior parts, and inserting the new Webster Sylphon Trap Attachments.

Re-tapping is not necessary for the new seat.

For conversion of Multiple-unit Water-seal Traps, see page 296.

Similar Webster Sylphon Attachments can be furnished for all the other sizes of Webster Water-seal Traps as follows:

¾-in.—No. 523
¾-in.—No. 533
1-in.—No. 534
1-in.—No. 544
1¼-in.—No. 545



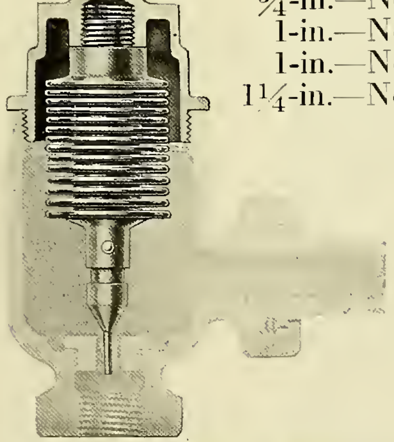
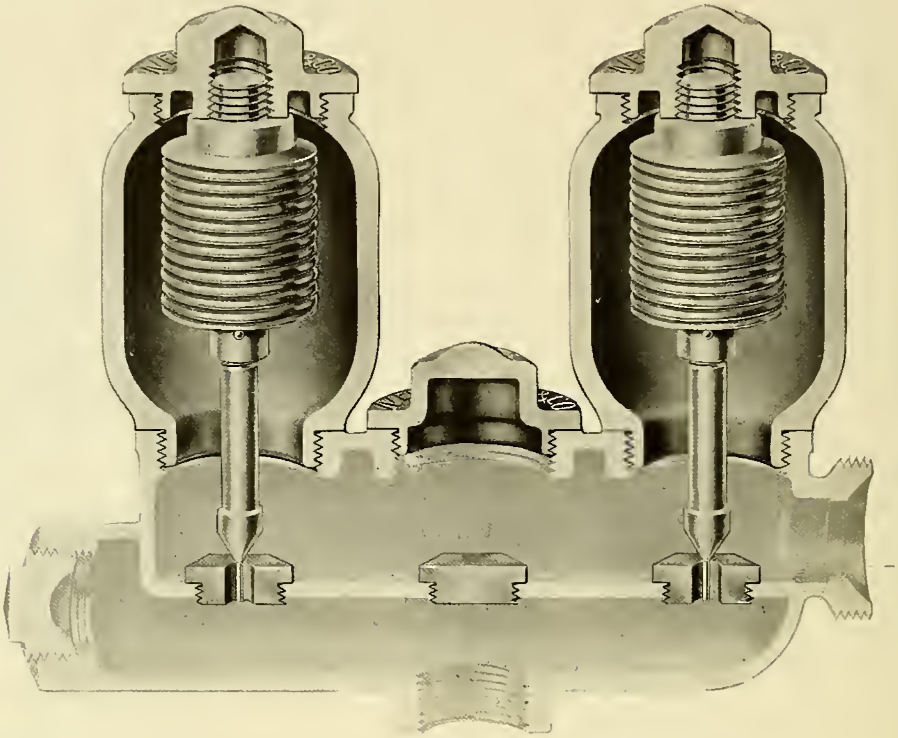
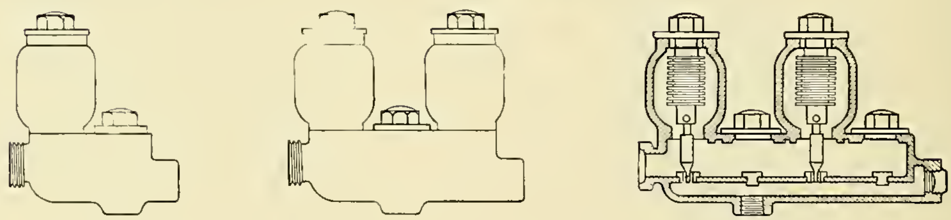


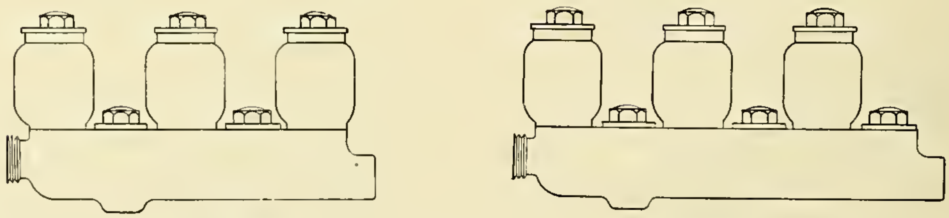
Fig. 26-5. No. 522 Webster Water-seal Trap in its original form and same trap changed over, using 522 Sylphon Attachment for higher discharge rating

The No. 522 and No. 523 take the same Sylphon Attachment. Another attachment applies equally for No. 533 and No. 534. No. 544 and No. 545 each have an individual attachment.





2-Unit



5-Unit 6-Unit

Fig. 26-6. Multiple-unit Thermostatic Valve with No. 4 Type bodies changed over by means of Webster Sylphon Attachments. Pipe connections untouched. Note how intervening openings are blanked out by new cap and solid seat

Conversion of Multiple-unit Webster Valves of Earlier Types: On units of radiation beyond the capacity of a single valve it was the practice

in the past to recommend and use a Multiple-unit Valve, made up of a special body having multiple openings to receive two or more bonnets similar in all respects to those used in the standard single-unit valve.

For changing these Multiple-unit Webster Valves by means of Sylphon Attachments, the use of Sylphon Attachments is recommended only for the alternate openings in the valve body, the intervening outlets being plugged as shown in Fig. 26-6.

Multiple Valves were made up to 6-unit. It is necessary to determine whether attachments are for 2-unit, 3-unit, etc., so that proper number of attachments, solid seats and blanking-out caps may be furnished.

The Multiple-unit Valve, when changed, will have capacity equal to (and possibly in excess of) the requirements of the original installation.

II. For "Sylphonizing" Radiator Outlet Valves of Other Makes

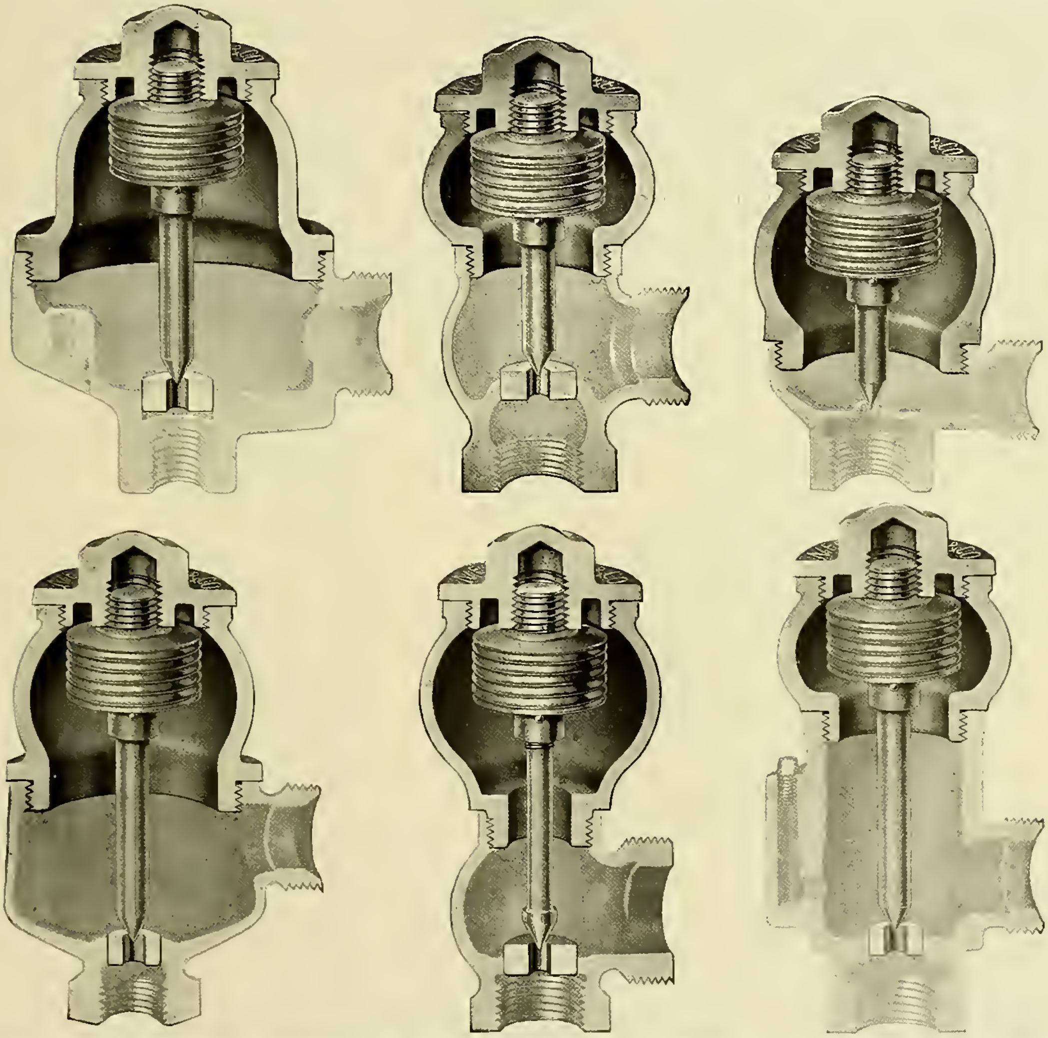


Fig. 26-7. 5-A Extension Attachments (Five-fold Sylphon bellows) applied to valve bodies of various makes

A great demand has developed for Webster Sylphon Attachments, not only in connection with early types of Webster Valves, but for other makes of valves and traps, and in the converting of old gravity systems in which the ordinary hand-wheel shut-off valve was employed.

To meet the requirements of a wide variety of sizes and types of valve and trap bodies the Attachments described in the following pages have been designed. The principle is the same with each attachment. The variation is only in the work of application.

With the instructions furnished and the tools loaned for the purpose, the work of Websterizing, by means of these attachments, is so simple that it can be done in a few minutes for each radiator, and so cleanly that there is no disturbance or damage to surroundings or furnishings.

The use of these Webster Sylphon Attachments, properly applied throughout the building, will often effect the same advantages as extensive changes in piping and at a small fraction of the cost. And further, the whole work of change-over can be done without interrupting the operation of the system as a whole.

Series 18 Webster Sylphon Attachments are of two general forms:

Class A in which the attachment parts are fitted in an extension body which screws into the old trap or valve body; and Class C in which the attachment parts are fitted into a special brass cap which is threaded to fit the old valve or trap body.

The Class A Extension Attachments are made with extension bodies to receive 5-fold Sylphon Bellows (symbol 5-A) and to receive 12-fold Sylphon Bellows (symbol 12-A).

The extension bodies of both the 5-A and 12-A classes are made with a

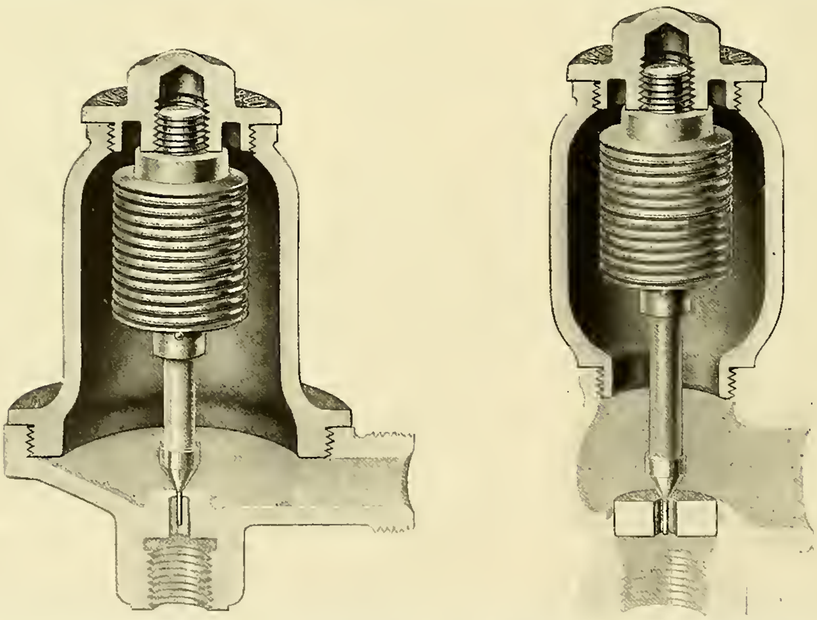


Fig. 26-8. Typical Class A Sylphon Attachments having extension bodies. Where necessary for securing correct final adjustment, a screw fit or push fit seat is used. A typical push fit seat is shown at the right.

5-A

Fig. 26-9. Typical Extension bodies 12-A

threaded opening at the top to receive a standard cap, but of varying diameters of the lower part of the body, so that the lower end may be threaded to fit the thread of the old body.

The illustrations show the full series of Extension Attachments from 5-A-12 to 5-A-27 inclusive. The 12-A Extension Attachments are similarly made in sizes 12-A-12 to 12-A-27 inclusive, although the application of only two of this type is shown.

The capacity required as indicated by size of radiator determines whether a 5-A or 12-A Extension Attachment should be used.

It will be noted that the valve stem attached to the Sylphon Bellows varies in length with the type of valve body, but is similar in all cases.

The seat requires a little explanation. It is impractical to use a threaded seat, as a constant distance must be maintained from body face to seat face and this cannot be done with a threaded seat because of the variations in the distance mentioned, which will occur in bodies of same make and size.

The seat is made to push-fit in the body opening which is previously prepared by reaming to the desirable diameter. Final attachment to gauge depth to meet any variation in the depth of the valve body is made by means of a push-in tool which is loaned for the purpose.

In the case of ordinary globe or angle valve bodies and in various makes of float traps where preparation in this respect was not previously provided, the push-fit seat described above provides means to obtain the correct final adjustment without difficulty.

The valve stem is a solid brass rod with a conical taper for seating and is of varying length as determined: (1) by the gauge depth of the old body from bonnet face to seat, (2) by the diameter of orifice in the seat; and (3) by the rating of the radiation unit to which the valve is connected. Where necessary to provide greater vapor space through the neck of the extension body, the rod is turned down to smaller diameter at such points.

The Class C Cap Sylphon Attachments are designed for those forms of old valve and trap bodies in which the expanding member (Sylphon Bellows) and conical valve piece may be placed entirely within the old body without the use of an extension body.

With this class of attachment it is necessary to provide a special cap, threaded to fit the existing body, but the design has been standardized so that few patterns need be used to meet a wide variety of bodies.

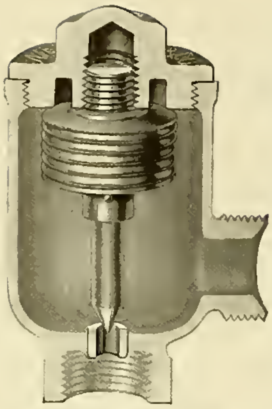
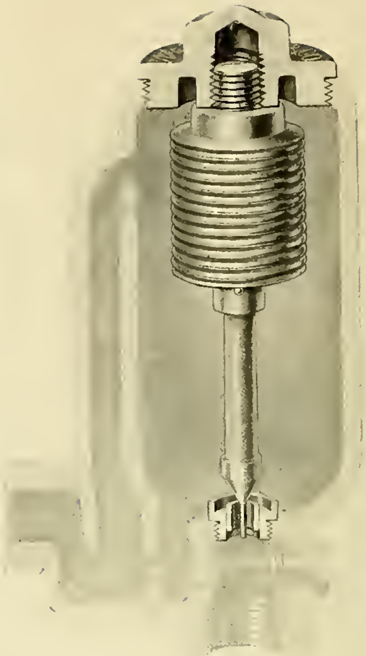


Fig. 26-10. Typical Class C Sylphon Cap Attachments placed entirely within the old bodies and push-fit seats installed for correct final adjustment

At the left is a 5-C Attachment (Five-fold Sylphon bellows)

At the right is a 12-C Attachment (Twelve-fold Sylphon bellows)

Note this special case of a new screwed-in seat with a pushed-in ferrule for insuring accurate adjustment



The Class C Cap Attachments, like the Extension Attachments, are made to receive either the 5-fold or 12-fold Sylphon Bellows to which the symbols 5-C and 12-C are given.

The illustrations above show the application of Class C Cap Attachments to two different shapes of valve bodies.

The description given previously in reference to the valve stem and seat for the Extension Attachments, applies equally to the Cap Attachments.

CHAPTER XXVII

Fuel Saving by Preheating Boiler-Feed Water

WHERE exhaust steam is available and would otherwise be wasted, a considerable saving of fuel may be effected by utilizing a direct-contact (open) feed-water heater to transfer heat from the exhaust steam to the cold feed water.

The saving amounts to approximately one per cent of fuel for each 11 deg. increase in the feed-water temperature. This is the figure taken for ordinary calculations.

A more accurate method of computing this saving takes into consideration the total heat in the steam generated in the boiler, as well as the final and initial temperatures of the feed water.

This formula is

Total saving in per cent $= \dfrac{100\,(t_1 - t_2)}{H + 32 - t_2}$, in which $H =$ total heat above 32 deg. fahr. per lb. of steam at boiler pressure, $t_1 =$ temperature of water after heating, and $t_2 =$ temperature of water before heating.

Table 27-1. Percentage of Total Heat of Steam Saved per Degree Increase in Feed-water Temperature for Various Pressures of Saturated Steam

Initial temp. Deg. fahr.	Gauge pressure in boiler—Lb. per sq. in.										
	0	10	25	50	75	100	125	150	175	200	225
	Value of H										
	1150.4	1160.2	1169.2	1178.4	1184.3	1188.8	1192.2	1195.0	1197.3	1199.2	1200.9
	Per cent saved per degree increase in temperature										
32	.0869	.0862	.0855	.0849	.0844	.0841	.0839	.0837	.0835	.0834	.0833
40	.0875	.0868	.0861	.0854	.0850	.0847	.0844	.0843	.0841	.0840	.0839
50	.0883	.0875	.0869	.0862	.0857	.0854	.0852	.0850	.0848	.0847	.0846
60	.0891	.0883	.0876	.0869	.0865	.0862	.0859	.0857	.0855	.0854	.0853
70	.0899	.0891	.0884	.0877	.0872	.0869	.0866	.0864	.0863	.0861	.0860
80	.0907	.0899	.0892	.0884	.0880	.0877	.0874	.0872	.0870	.0869	.0867
90	.0915	.0907	.0900	.0892	.0888	.0884	.0882	.0879	.0878	.0876	.0875
100	.0924	.0916	.0908	.0900	.0896	.0892	.0889	.0887	.0886	.0884	.0883
110	.0932	.0924	.0916	.0909	.0904	.0900	.0897	.0895	.0893	.0892	.0891
120	.0941	.0933	.0925	.0917	.0912	.0909	.0906	.0903	.0902	.0900	.0899
130	.0950	.0941	.0934	.0925	.0921	.0917	.0914	.0912	.0910	.0908	.0907
140	.0959	.0950	.0942	.0934	.0929	.0925	.0922	.0920	.0918	.0916	.0915
150	.0968	.0959	.0951	.0943	.0938	.0935	.0931	.0929	.0927	.0925	.0924
160	.0978	.0969	.0960	.0952	.0947	.0943	.0940	.0937	.0935	.0934	.0932
170	.0987	.0978	.0970	.0961	.0956	.0952	.0948	.0946	.0944	.0942	.0941
180	.0997	.0988	.0979	.0970	.0965	.0961	.0957	.0955	.0953	.0951	.0950
190	0.1008	.0998	.0989	.0980	.0974	.0970	.0967	.0964	.0962	.0960	.0959
200	0.1018	0.1008	.0999	.0990	.0984	.0980	.0976	.0974	.0972	.0970	.0968
210	0.1028	0.1018	0.1009	.0999	.0994	.0990	.0986	.0983	.0981	.0979	.0978
220	0.1039	0.1029	0.1019	0.1010	0.1004	.0999	.0996	.0993	.0991	.0989	.0987

Example: Assume a boiler pressure of 140 lb. per sq. in. absolute, and initial and final temperatures of 40 deg. fahr. and 210 deg. fahr. respectively. The total saving according to this formula is 14.36 per cent, where by the "one per cent for each 11-deg. increase" rule, the saving for the same conditions figures 15.45 per cent.

For convenience the results as figured from the more accurate formula have been reduced in Table 27-1, to a basis of per cent of saving per degree increase of temperature.

Webster Feed-water Heaters: Webster Feed-water Heaters, for obtaining the fuel savings just mentioned and other benefits not so easily measured, are made in the following types:

Series 100, Class B, with overflow seal: The standard type for utilizing

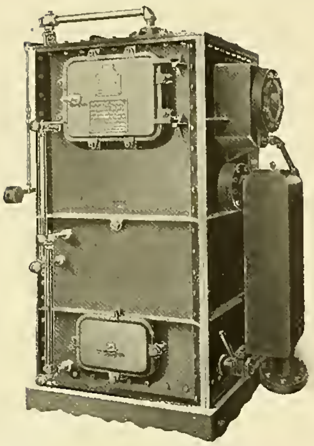


Fig. 27-1. Series 100 Class B Webster Feed-water Heater

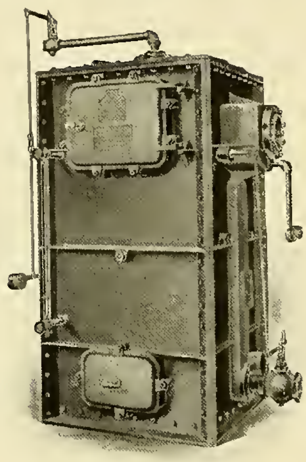


Fig. 27-2. Series 200 Class EB and Series 300 Class EBH Webster Feed-water Heater. Standard Type. Smaller sizes

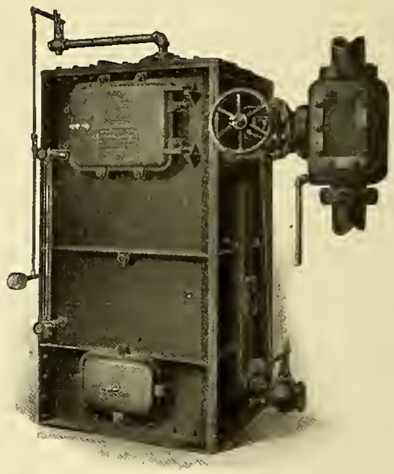


Fig. 27-3. Series 400 Class EBP and Series 500 Class EBPH Webster Feed-water Heater. Preference Cut-out Type

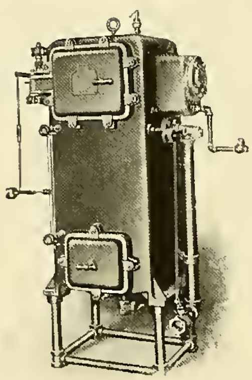


Fig. 27-4. Series 800 Class EF Webster Feed-water Heater, Standard Type

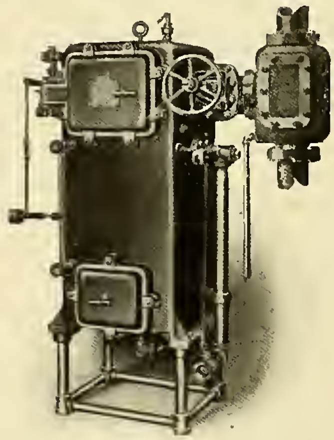


Fig. 27-5. Series 900 Class EFP Webster Feed-water Heater. Preference Cut-out Type

exhaust steam at atmospheric pressure and for a maximum steam pressure of ½-lb. per sq. in. May be operated on either induction or thoroughfare principle.

Series 200, Class EB: The standard type for use in connection with exhaust steam systems under pressures not exceeding 5-lb. per sq. in. Best operated on induction principle.

Series 300, Class EBH: Same as Series 200, Class EB, but suitable for pressures up to 10-lb. per sq. in. maximum. Tested to 15-lb. per sq. in.

Series 400, Class EBP: Same as Series 200, Class EB, but with independent oil separator large enough to purify all exhaust. Specially designed for use with exhaust steam heating or drying systems under pressures not exceeding 5-lb. per sq. in.

Series 500, Class EBPH: Same as Series 400, Class EBP, but suitable for pressures up to 10-lb. per sq. in. maximum. Tested to 15-lb. per sq. in.

Series 800, Class EF: This type is for smaller capacities, 50 to 350 hp., and is similar to Series 200, Class EB, except that the shell is a one-

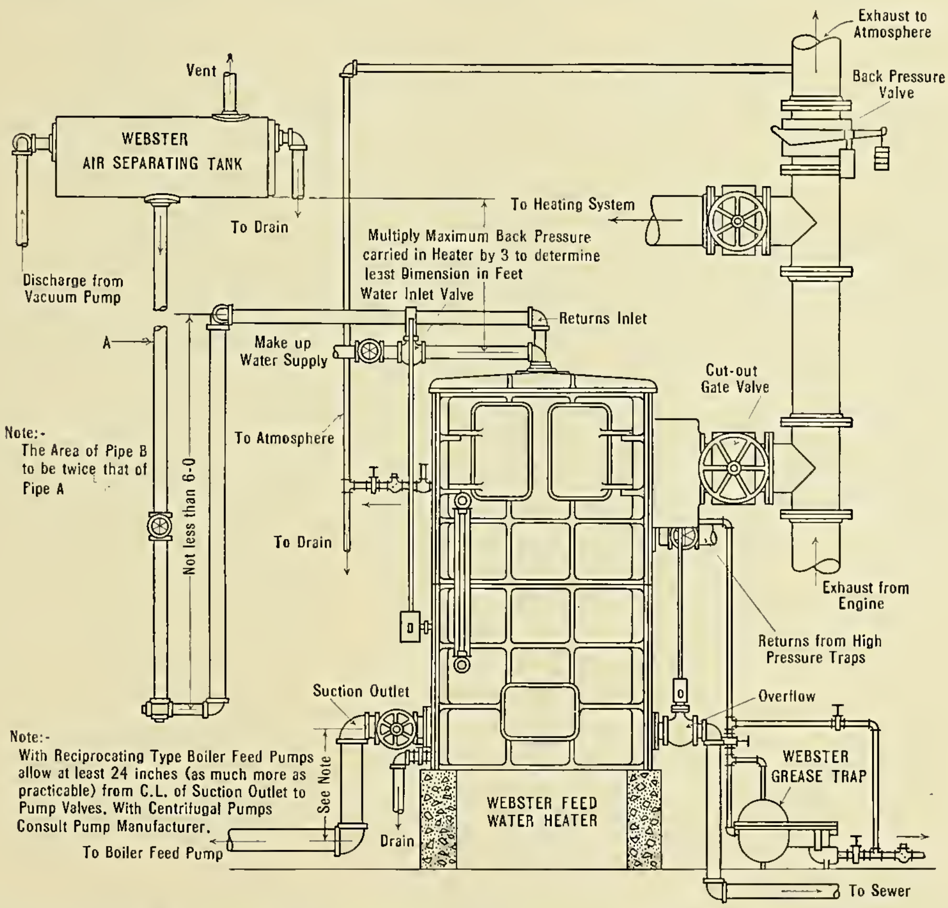


Fig. 27-6. Webster Feed-water Heater installation in connection with a Vacuum Heating System. Water inlet automatically controlled. The heater shown is of the standard type. Any other type of Webster Heater would be connected in the same way

piece casting and is supported by a framework made from pipe and fittings. It is suitable for working pressures up to 10-lb. per sq. in.

Series 900, Class EFP: Same as Series 800, Class EF, but including the large size oil separator and the cut-out valve.

Webster Feed-water Heaters, Standard Type: The heater shell as illustrated in this chapter, is made of close-grained cast-iron plates. Webster Heaters are also made with shells of genuine old-fashioned puddled wrought-iron, or of other sheet metals such as flange steel or the so-called copper-bearing steels. Wrought-iron heaters are specially recommended as they are proof against the minor accidents of operation which frequently crack cast-iron heaters.

The heater is easily cleaned, as the interior is accessible without disturbing any of the pipe connections. The large hinged doors may be quickly opened, and the trays withdrawn. The lower chamber, containing the

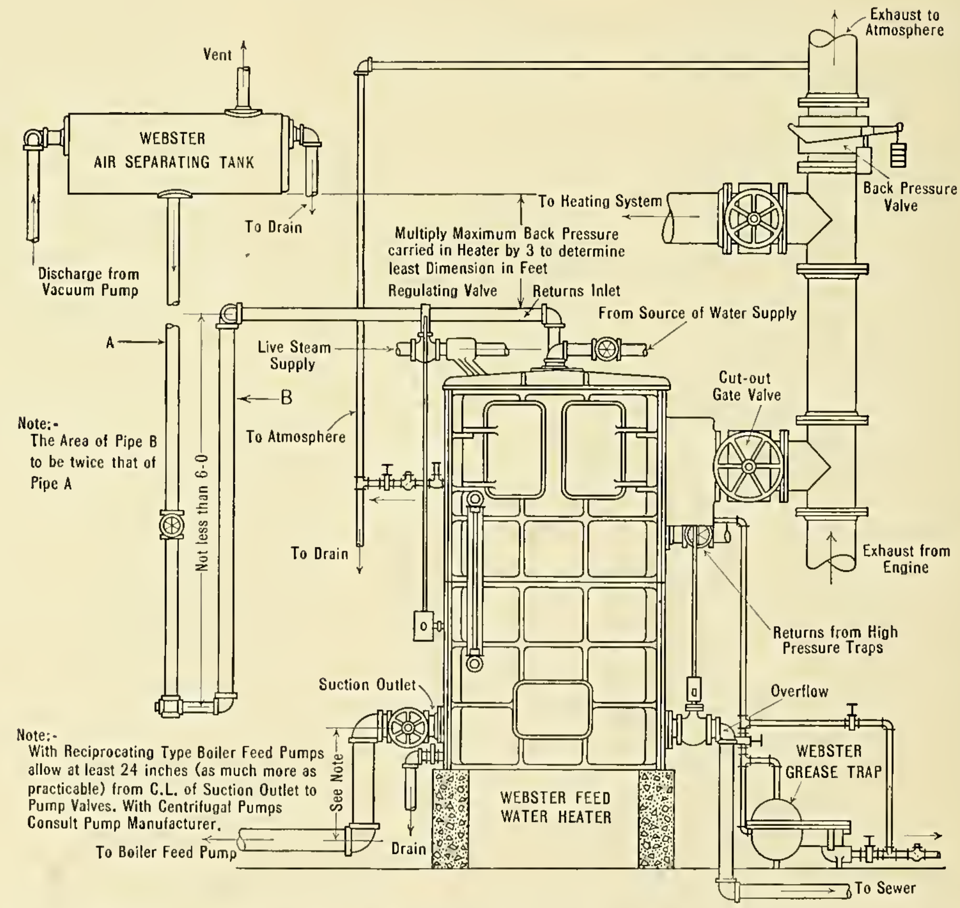


Fig. 27-7. This Webster Feed-water Heater installation differs from usual practice in that the make-up water supply is manually controlled. A float within the heater operates a valve in the steam-pipe supplying the boiler-feed pump to stop the pump when the water level is below a pre-determined point

filter, is accessible through the filter doors. Where the doors are bolted to the heater body, the shell is suitably reinforced, the faces being machined to insure tight joints.

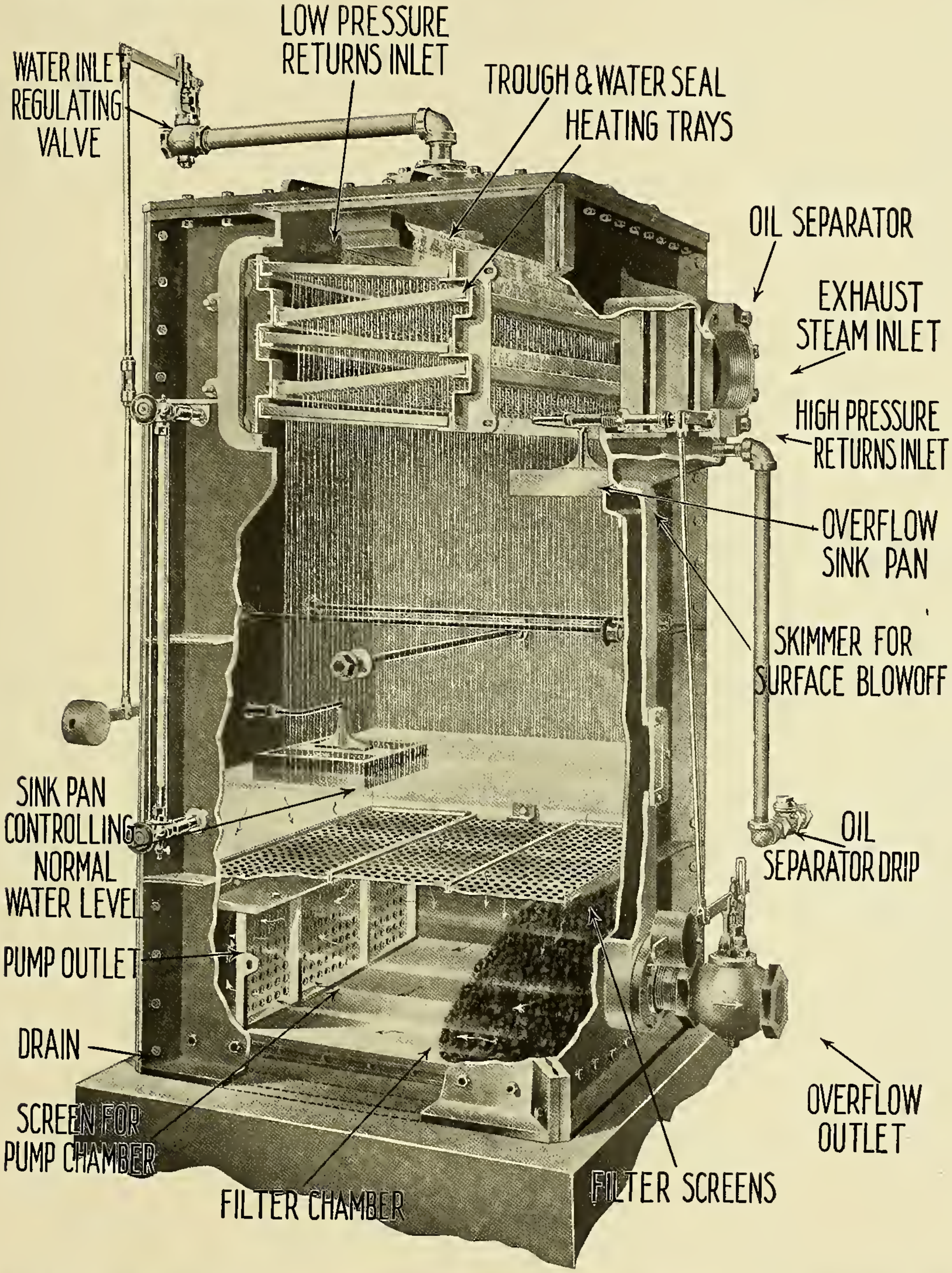


Fig. 27-8. Series 200 Class EB and Series 300 Class EBH Webster Feed-water Heater, Standard Type

The water supply to the heater is controlled automatically, the regulating valve being operated by a series of levers connected to an open copper sink pan (performing the functions of a float), placed within the heater shell.

Any dangerous excess of water automatically passes out of the heater when the water reaches the overflow level. Except in the case of the 100 Series, the excess water is automatically passed out through a valve actuated in the same manner as the cold water supply-valve, that is, by another open sink pan placed within the heating chamber. This valve is normally closed, preventing loss of steam.

The Webster Oil Separator which forms a part of each heater is well

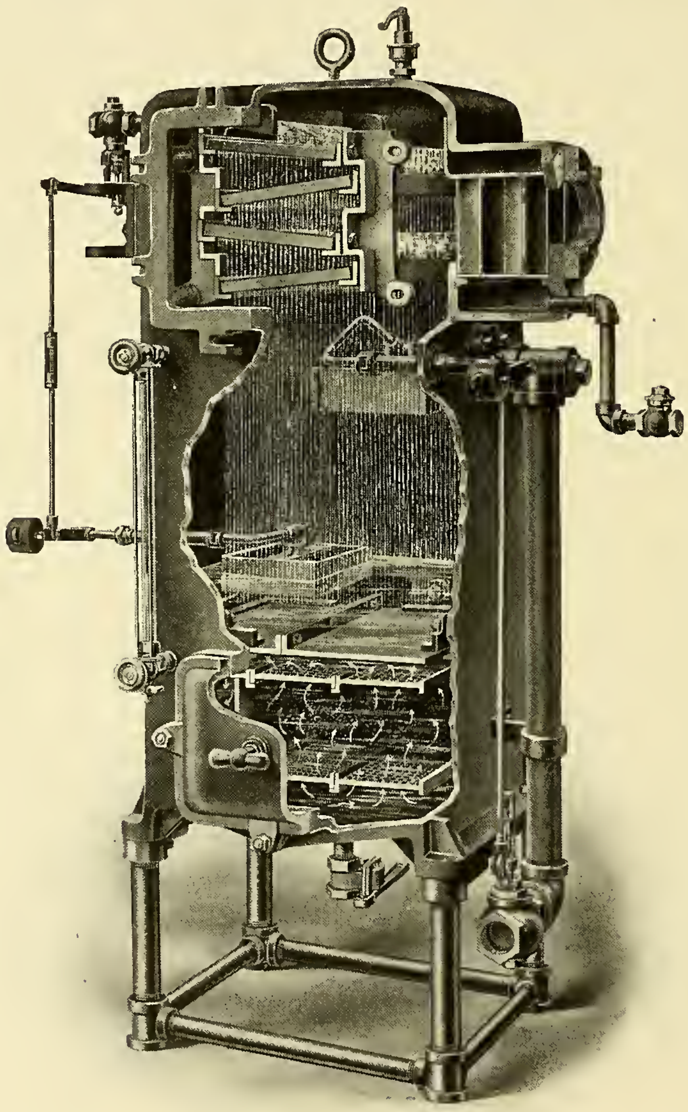


Fig. 27-9. Series 800 Class EF Webster Feed-water Heater, Standard Type

known and extensively used as an independent unit for removing oil from exhaust steam mains, hence its use in the Webster Feed-water Heater.

The feed water, entering the heater through the automatically controlled valve inlet, passes into the water-sealed distributing trough, which has two wide, extended lips. The water, overflowing from this trough in even sheets, is distributed over a series of oppositely inclined, finely perforated metal trays, arranged one above the other as shown in the illustration below. The water in its downward course falls from one tray to the other, part of it passing through the tray perforations and the balance

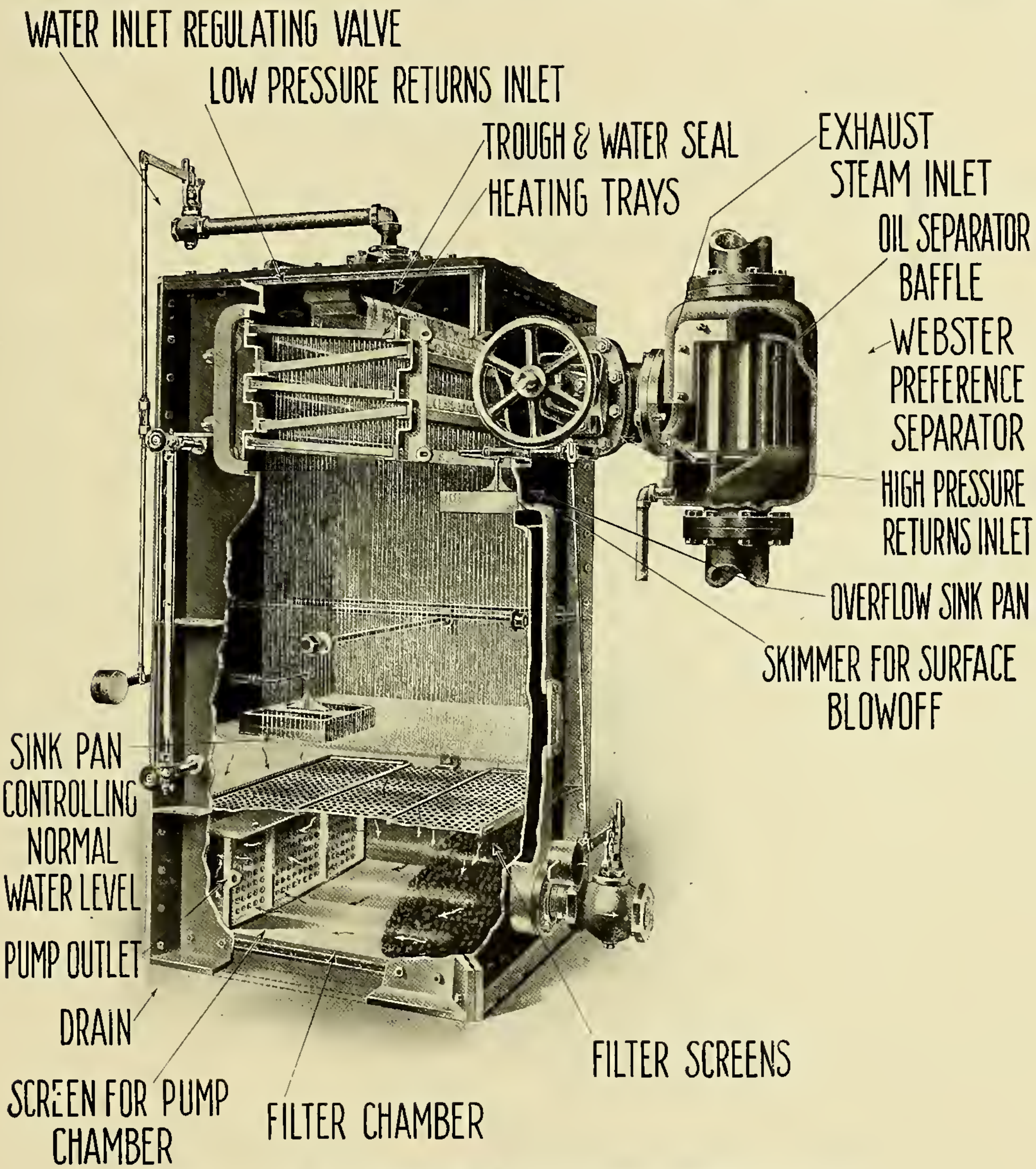


Fig. 27-10. Series 400 Class EBP and Series 500 Class EBPH Preference Cut-out Type Webster Feed-water Heaters

falling from the lower edge of the tray to the tray immediately below.

This method of water travel provides the necessary surface contact for the steam and water so that the highest possible temperature is imparted to the water, causing a liberation of gases and precipitation of solids. Ample space is provided for uniform distribution of steam around the trays.

By reason of the large storage chamber it is possible to utilize the heater as a receiver for condensation from heating systems, dry kilns, heating apparatus, etc. Between the level at which the cold water supply-valve is closed and the overflow there is ample space for the accumulation and storage of such returns.

The filter is located in the lower compartment of the heater. In this settling chamber, opportunity is given for the precipitation and filtration of the particles of sediment and impurities and for frequent drainage through a quick-opening drain valve.

The filter bed is commonly composed of coke or other suitable material, which is contained between the perforated division screens already mentioned. This material can be renewed whenever necessary.

The large doors at the front allow ready access for charging and cleaning.

The Webster Preference Cut-out Heater: This type, as may be noted from the illustrations, combines a Webster Heater and a large oil separator with a cut-out gate valve intervening. The oil separator has sufficient capacity to remove the oil from the exhaust steam delivered from the engines, pumps and other sources. This arrangement is therefore especially desirable where exhaust steam is to be utilized in heating or drying systems, cooking kettles or other industrial processes.

A Webster Grease Trap is used in draining the separator. Steam from the engines and auxiliaries should be combined in a common exhaust pipe before reaching the heater. This exhaust pipe may enter the separator horizontally or vertically, the latter condition being usual with the exhaust steam current upward.

Upon reaching the preference oil separator the steam flows horizontally through the baffles, which are of the standard Webster design (see Figure 27–11), comprising a number of hooked steel plates interposed in the course of the steam, causing separation by contact, by change of direction and by adhesion. The ports through which the steam is guided and the free area through the baffles are especially designed to prevent any considerable loss of pressure.

After passing through the baffles, the steam may pass to the heater, or to the outlet into the heating system or other apparatus using exhaust steam or to the atmosphere.

Particularly valuable advantages of the Webster Preference Cut-out Heater are:

1. The considerable saving in piping connections and additional apparatus accomplished by its use as compared with the Standard Heater.

2. The cut-out valve used in the Webster Preference Cut-out Heater is most reliable for its purpose. When the heater is cut out for internal inspection or cleaning, the course of the exhaust steam through the oil

separator is such that no steam is in contact with the side wall of the heater. Steam passes through the separator and on to atmosphere or the heating system without warming up the heater body to a degree that would endanger or discomfort the man who may have to enter. A thorough clean-out is possible at any time without having to wait until the whole plant is shut down.

3. The grease and oil trap too is not integral with the overflow of the heater, so that if its outlet becomes temporarily deranged, oil cannot get back into the heater through the overflow opening.

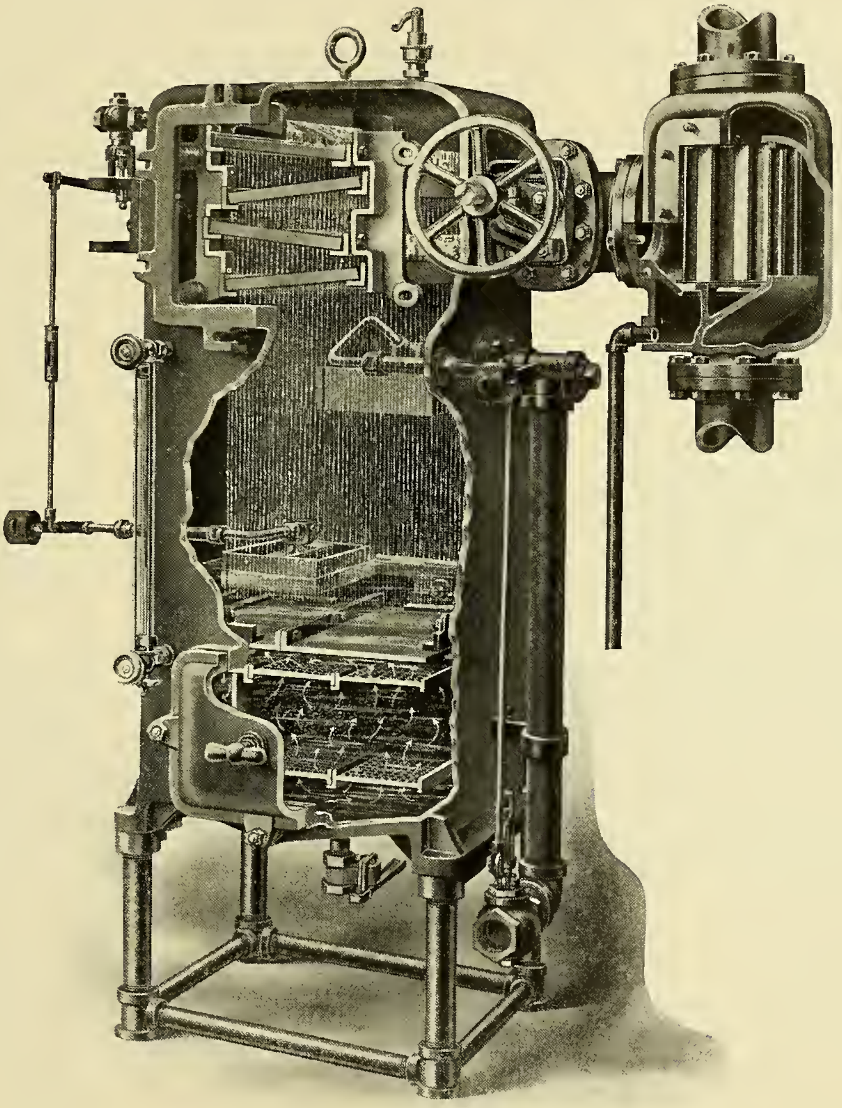


Fig. 27-11. Series 900 Class EFP Preference Cut-out Type Webster Feed-water Heater

Table 27-2. Dimensions of Series 200, Class EB, Webster Feed-water Heaters

For working pressure up to 5 lb. per sq. in.

Specifications

No.	Capacity		Drawing no.	Heating trays		Cubic contents		Filter	Wkg. pres.	Weights, lb.	
	Horsepower *	Lb. min.		Area sq. ft.	Material	Total cu. ft.	Water cu. ft.			Shipping	Max.
203	to 400	No. of lb. per min. = ½ of rated horsepower	9247	12.5	American ingot iron or copper	24.4	14.7	Downward flow	5 lb. per sq. in.	2600	3600
205	425 to 650		9203	16.5		40.0	25.5			3700	5400
207	675 to 900		9250	24.0		60.0	40.0			4700	7300
210	925 to 1350		9254	33.0		80.5	52.0			5700	9000
215	1375 to 1850		9252	51.6		121.3	80.0			8000	13100
220	1875 to 2400		9257	63.8		152.5	104.0			9000	15700
225	2425 to 3000		9256	82.0		180.0	128.1			10300	18400
230	3100 to 4000		22457	95.7		240.0	133.5			13000	21300
235	4100 to 5500		13377	121.5		316.0	140.0			15000	23600
250	5600 to 7500		13626	160.1		400.0	179.0			20000	31400
285	7600 to 9500		22196	201.5		482.0	222.0			22000	36000
299	9600 to 12000		18779	243.0			268.0			25000	41700

* One rated horsepower=capacity for heating 30 lb. of water per hour from 40 deg. fahr. to a temperature within 5 deg. of the steam temperature

No.	Connections								
	①	②	③	④	⑤	⑥	⑦	⑧	⑨
203	6	1	4	2½	¾	1½	2½	2½	1
205	8	1½	4	3	1	1½	3	2½	1
207	8	2	5	3½	1	2	4	3	1
210	10	2	5	3½	1	2	5	3	1
215	12	2½	6	4	1	2½	5	3	1¼
220	14	2½	6	4	1¼	3	5	5	1½
225	16	3	8	5	1¼	4	5	5	1½
230	18	3	8	6	1¼	4	2–5	5	1½
235	20	4	10	8	1¼	5	2–3	5	1½
250	22	5	10	8	1¼	5	2–3	5	1½
285	24	5	12	8	2–1½	5	2–4	6	2
299	28	6	12	10	2–1½	5	2–6	8	2

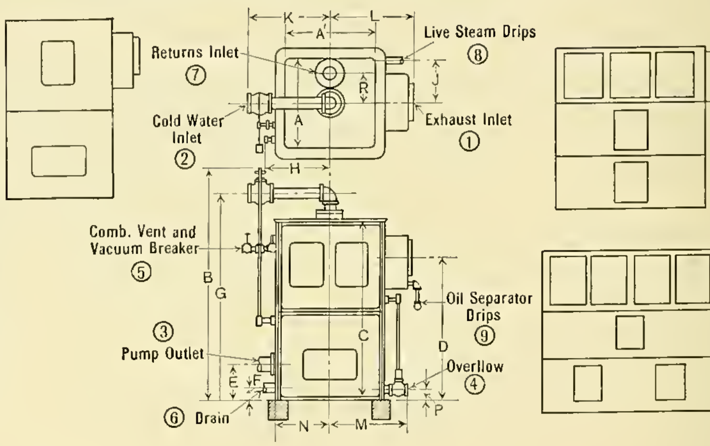


Fig. 27-12

No.	Trays		Foundation		Overall	Dimensions															
	No.	Size	Lg.	Wd.	Hgt.	A	A'	B	C	D	E	F	G	H	J	K	L	M	N	P	R
203	5	15 x24	35	35	80¾	26	26	80¾	66	54⅝	6¾	4⅛	73¼	18½	9	21½	21¾	25¼	16	5¾	9
205	5	15½x30⅝	41	41	88	32	32	88	72	57⅝	7¼	5¼	79¾	21½	11½	27	25½	28¾	19⅛	7¼	8¾
207	6	16 x36	45	45	101⅝	36	36	101⅝	84	69¼	7¼	6	93¼	25	13¾	28½	27½	34	21⅛	9⅛	11
210	12	10 x40⅜	51	51	101⅝	42	42	101⅝	84	67¼	7⅞	6	93¼	28	15½	31½	31½	37	24⅝	8	10½
215	12	13½x46	57	57	115⅛	48	48	115⅛	96	77⅛	8½	7	104¾	33¼	16	36	36¾	41½	27⅝	8¼	13¾
220	12	16¾x47	69	57	115½	48	60	115½	96	81⅝	8½	7	105½	40	18	41½	45	47½	33⅝	10¾	13¾
225	24	17½x28	69	66	117¾	57	60	117¾	96	82⅜	9	7½	106½	40¾	19¾	42	42⅝	46¼	33⅝	10¾	16
230	18	16¾x47	93	57	115¼	48	84	116¼	96	77	9	7½	105⅛	52	20	57	55	53⅞	45⅝	11¼	12½
235	24	15½x47	105	57	120⅝	48	96	120⅝	96	77	12½	9¼	105⅞	61¼	23¾	64	61	63⅛	51⅝	12	13¾
250	48	15½x31	105	72	122⅞	63	96	122⅞	96	75	11½	9¼	107⅜	61¾	25⅞	65	67¼	63⅛	51⅝	12	20
285	48	15½x39	105	89	122⅜	80	96	122⅜	96	75	....	9¼	107⅜	61¼	27	65	67¼	63⅛	....	12	16
299	48	15½x47	105	105	124⅜	96	96	124⅜	96	75	....	8½	107⅜	61¼	40	65	67¼	67⅛	....	12	36½

All sizes and dimensions in inches

NOTE: The above data (except weights) applies also to Extra-heavy 300 Series Class EBH Heaters for working pressures up to 10 lb. per sq. in.

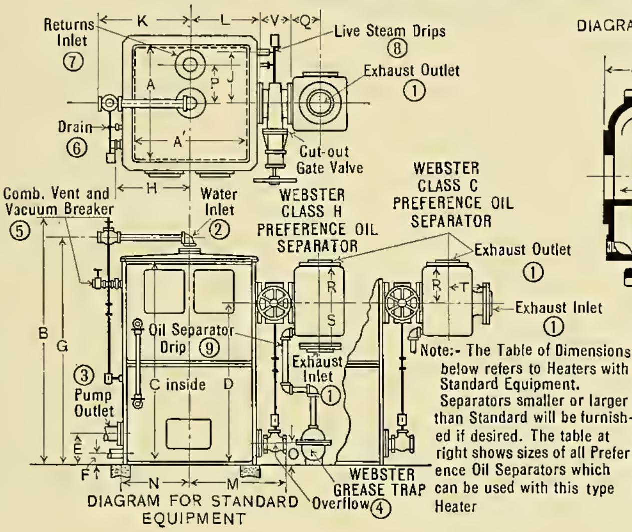
Table 27-3. Dimensions of 400 Series Class EBP Webster Feed-water Heaters

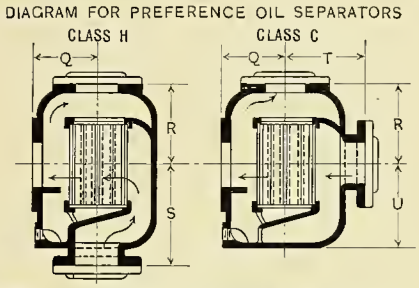
For working pressure up to 5 lb. per sq. in.

Specifications

No.	Capacity		Drawing no.	Heating trays		Cubic contents		Filter	Wkg. pres.	Weights, lb.	
	Horsepower *	Lb. min.		Area sq. ft.	Material	Total cu. ft.	Water cu. ft.			Shipping	Max.
403	to 400	No. of lb. per min. = ½ of rated horsepower	13166	12.5	American ingot iron or copper	24.1	14.7	Downward flow	5 lb. per sq. in.	3500	4500
405	425 to 650		13188	16.5		40.0	25.5			4950	6650
407	675 to 900		13167	24.0		60.0	40.0			6700	9300
410	925 to 1350		13165	33.0		80.5	52.0			8050	11350
415	1375 to 1850		13171G	51.6		121.3	80.0			10800	15900

* One rated horsepower = capacity for heating 30 lb. per hour from 40 deg. fahr. to a temperature within 5 deg. of the steam temperature





SIZE	CAPACITY LBS. STEAM PER MIN.	DIMENSIONS					SIZE DRIP
		Q	R	S	T	U	
6	46	8	11	13	9⅝	10¾	1
8	80	9¼	12¼	14	11	11⅝	1
10	125	11¾	14⅜	17⅞	14¾	14	1
12	175	13¼	15½	18	15⅜	15¼	1
14	255	15	18	23	19⅛	18⅛	1½
16	335	17¾	21	27¼	22¼	22	1½
18	375	19¾	24	29⅜	24½	23⅝	1½
20	475	20¼	25¾	30¾	25¼	24¾	1½

Fig. 27-13

Size no.	Standard Equipment			Connections									Trays		Foundation		Overall height
	Size sep'r	Size valve	Size trap	①	②	③	④	⑤	⑥	⑦	⑧	⑨	No.	Size	L'gth	Width	
403	10	6	1	10	1	4	2½	¾	1½	2½	2½	1	5	15 x24	35	35	80¾
405	12	8	1	12	1½	4	3	1	1½	3	2½	1	5	15½x30⅝	41	41	88
407	16	8	1½	16	2	5	3½	1	2	4	3	1	6	16 x36	45	45	101⅝
410	18	10	1½	18	2	5	3½	1	2	5	3	1	12	10 x40⅜	51	51	101⅝
415	20	12	1½	20	2½	6	4	1	2½	5	3	1¼	12	13½x46	57	57	115⅛

Size no.	Dimensions																
	A	A'	B	C	D	E	F	G	H	J	K	L	M	N	O	P	V
403	26	26	80¾	66	54⅝	6¾	4⅜	73¼	18½	9	21¾	17½	25¼	16	5¾	9	10½
405	32	32	88	72	57⅝	7¼	5¼	79¾	21½	11½	27	20½	28¾	19⅛	7¼	8¾	11½
407	36	36	101⅝	84	69¼	7¼	6	93¼	25	13¾	28½	22½	34	21⅛	9⅛	11	11½
410	42	42	101⅝	84	67¼	7⅞	6	93¼	28	15½	31½	25½	37	24⅝	8	10½	13
415	48	48	115⅛	96	77⅛	8½	7	104¾	33¼	16	36	28½	41½	27⅝	8¼	13¾	14

Note: The dimensions and data above, except weights, may be used also for the 500 Series Class EBPH Extra-heavy Pattern Webster Feed-water Heaters

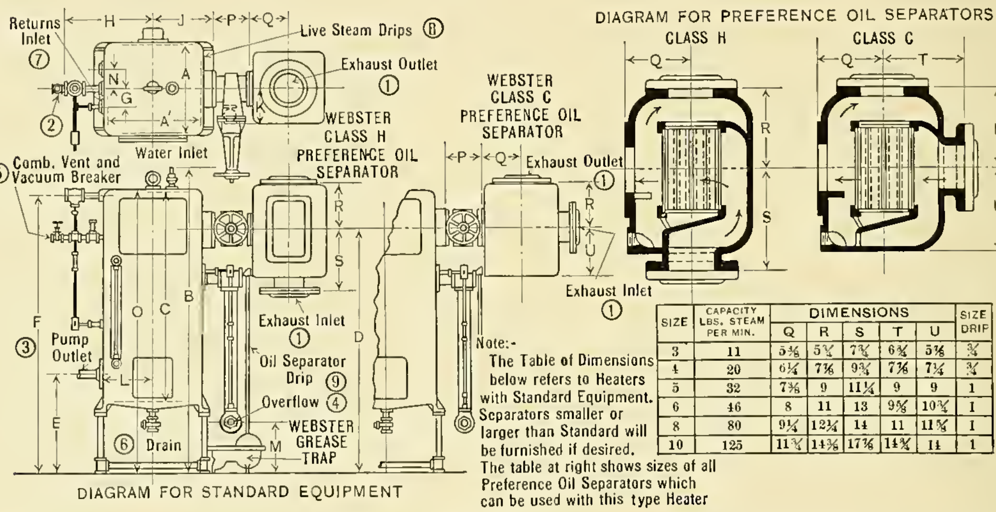
Table 27-4. Dimensions of 900 Series Class EFP Webster Feed-water Heaters

For working pressure up to 10 lb. per sq. in.

Specifications

No.	Capacity: Horsepower*	Capacity: Lb. min.	Drawing no.	Heating trays: Area sq. ft.	Heating trays: Material	Cubic contents: Total cu. ft.	Cubic contents: Water cu. ft.	Filter	Wkg. pres.	Weights, lb.: Shipping	Weights, lb.: Max.
900	to 90	No. of lb. per min. = ½ of rated horsepower	17198	4.5	American ingot iron or copper	7.1	4.2	Upward flow	10 lb. per sq. in.	1675	1925
901	95 to 150		16837	5.0		9.8	5.9			1780	2140
901½	155 to 225		16724	5.6		11.6	7.3			2200	2600
902	230 to 300		17203	9.0		16.4	11.08			2700	3425

* One rated horsepower = capacity for heating 30 lb. per hour from 40 deg. fahr. to a temperature within 5 deg. of the steam temperature



SIZE	CAPACITY LBS. STEAM PER MIN.	DIMENSIONS: Q	R	S	T	U	SIZE DRIP
3	11	5⅜	5¼	7¾	6¾	5⅜	¾
4	20	6¼	7⅜	9¾	7⅞	7¼	¾
5	32	7⅜	9	11¼	9	9	1
6	46	8	11	13	9⅝	10¾	1
8	80	9¼	12¼	14	11	11⅝	1
10	125	11¾	14⅜	17⅞	14¾	14	1

Fig. 27-14

Size no.	Standard equipment: Size sep'r	Size valve	Size trap	Connections: ①	②	③	④	⑤	⑥	⑦	⑧	⑨	Trays: No.	Trays: Size	Foundation: L'gth	Width	Overall height
900	4	3	¾	4	1	2	1½	¾	1¼	1½	1¼	¾	4	10x16	26	23	62
901	6	4	1	6	1	2½	2	¾	1¼	1½	1½	1	4	10x18	28	25	68½
901½	8	4	1	8	1	3	2	¾	1¼	1½	1½	1	4	10x20	28	27	72½
902	8	5	1	8	1	3	2	¾	1¼	2	1½	1	4	14x23	30	30	79

Size no.	Dimensions: A	A'	B	C	D	E	F	G	H	J	K	L	M	N	O	P
900	16	18	62	43¼	48	20½	55¼	3¼	18½	14	7⅛	9¾	10⅛	3¼	57	8
901	18	20	68½	47¼	54⅜	23	61¾	3⅞	19½	13	9⅜	10¾	11⅜	3⅞	63½	9
901½	20	20	72½	51	58⅛	23	65½	4⅞	19½	13½	9⅜	10¾	11⅜	4⅞	67¼	9
902	22¾	22¾	79	56½	63⅝	24½	71⅝	5	21	13¼	9½	12¼	12⅞	5	73½	10

All sizes and dimensions in inches

Table 27-5. Dimensions of Series 800, Class EF, Webster Feed-water Heaters

For working pressure up to 10 lb. per sq. in.
Specifications

No.	Capacity		Drawing no.	Heating trays		Cubic contents		Filter	Wkg. pres.	Weights, lb.	
	Horsepower *	Lb. min.		Area sq. ft.	Material	Total cu. ft.	Water cu. ft.			Shipping	Max.
800	to 90	No. of lb. per min. = ½ of rated horsepower	17045	4.5	American ingot iron or copper	7.1	4.2	Upward flow	10 lb. per sq. in.	1125	1400
801	95 to 150		16660	5.0		9.8	5.9			1450	1850
801½	155 to 225		16661	5.6		11.6	7.3			1700	2200
802	230 to 300		16662	9.0		16.4	11.08			2200	2900

* One rated horsepower = capacity for heating 30 lb. of water per hour from 40 deg. fahr. to a temperature within 5 deg. of the steam temperature

No.	Connections								
	1	2	3	4	5	6	7	8	9
800	3	1	2	1½	¾	1¼	1½	1¼	¾
801	4	1	2½	2	¾	1¼	1½	1½	¾
801½	4	1	3	2	¾	1¼	1½	1½	¾
802	5	1	3	2	¾	1¼	2	1½	¾

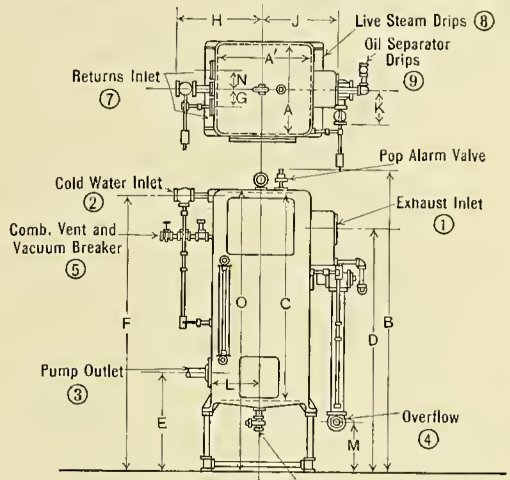


Fig. 27-15

No.	Trays		Water line			Filter			Dimensions															
	No.	Size	O'er	Pow.	Rec.	Th.	Ar.	Cu. ft.	A	A'	B	C	D	E	F	G	H	J	K	L	M	N	O
800	4	10x16	39⅛	35½	32⅝	6	2.0	.9	16	18	62	43¼	48	20½	55¼	3¼	18½	14	7⅛	9¾	10⅛	3¼	57
801	4	10x18	44¾	38¾	35⅛	6	2.5	1.2	18	20	68½	47¼	54⅜	23	61¾	3⅞	19½	17⅛	9⅜	10¾	11⅜	3⅞	63½
801½	4	10x20	48½	42½	35⅛	6	2.8	1.4	20	20	72½	51	58⅛	23	65½	4⅞	19½	17⅛	9⅜	10¾	11⅜	4⅞	73½
802	4	14x23	55½	49¼	36⅝	6	3.6	1.8	22¾	22¾	79	56½	63⅝	24½	71⅝	5	21	19¼	9½	12¼	12⅞	5	74

All sizes and dimensions in inches

THE WEBSTER-LEA HEATER METER: This apparatus combines the Webster Feed-water Heater of the rectangular cast-iron type, with the Lea V-Notch Recording Meter so arranged that both may be operated in combination or either independently of the other.

Besides heating the boiler feed water to the boiling point, this apparatus indicates the actual amount of boiler evaporation. Its continuous meter records show up careless or improper firing methods, leakage, condensation due to poor installation, inferior coal and in other words, act as a check upon the general efficiency of the entire boiler plant.

The charts (Fig. 27–17) can be integrated by means of a standard planimeter, and an integrating attachment giving the total flow for any period is supplied. The readings from the integrating attachment indicate approximately quantities of water which have passed over the weir.

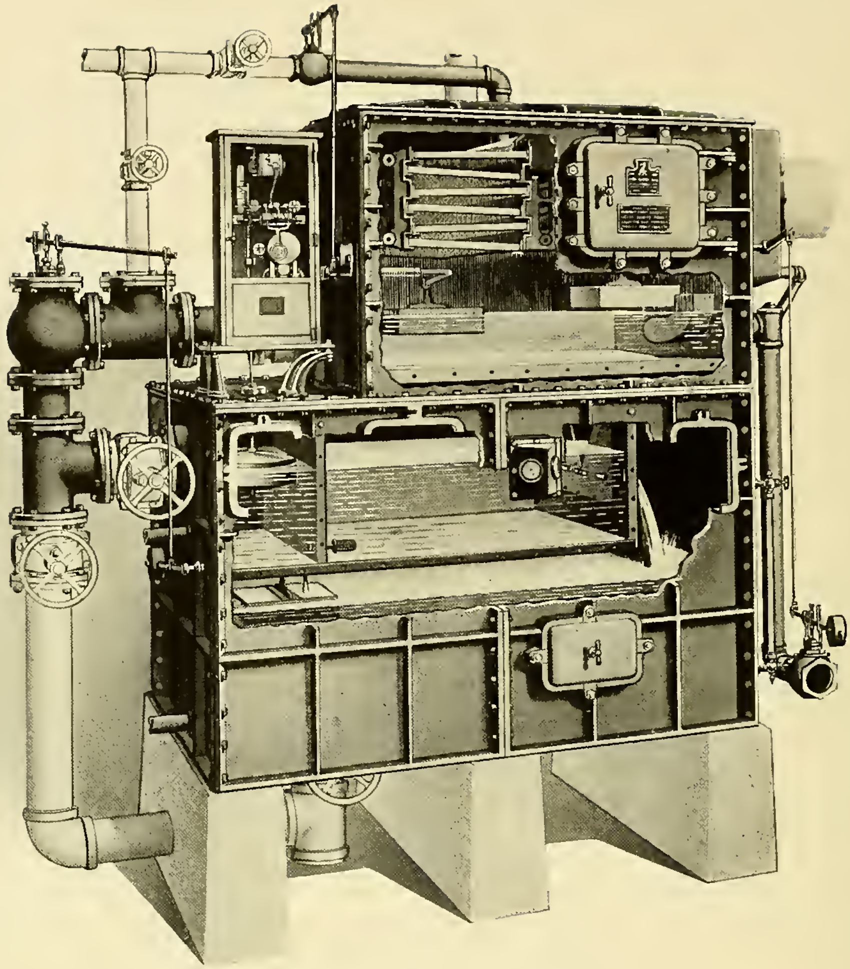


Fig. 27-16. Typical Webster-Lea Heater Meter

Where it is desired to have a record of the feed-water temperature on the same chart with the meter record, a special attachment can be fitted to any standard instrument. The meter chart and drum are made wider to provide 2½ inches for temperature calibrations. This space has 25 equal divisions calibrated in any specified 50 or 100-deg. interval.

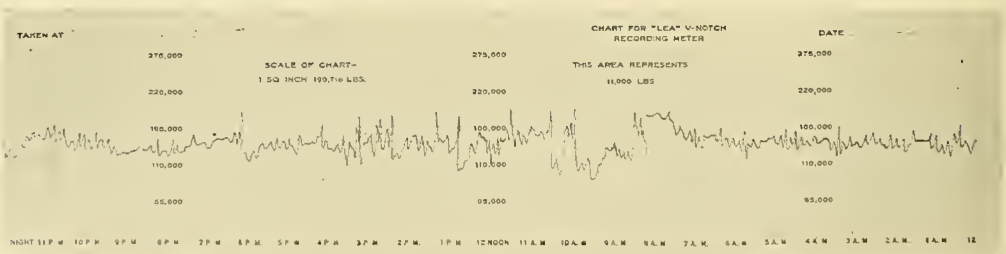


Fig. 27-17. Typical chart from a Webster-Lea Heater Meter

Part III—Addenda

CHAPTER XXVIII

Miscellaneous Useful Information

THE tables in the following pages cover many subjects for which the Heating Engineer must have readily available data. They have been selected after careful consideration and will be found reliable and sufficiently accurate in every respect to meet the requirements of good practice.

The tables on any subject can be readily located by reference to the back of the book, where they are included both in the general index and the special index of tables.

Table 28-1. Diameters and Weights of Seamless Brass and Copper Tubes*

Iron Pipe Size and Plumber's Size

Iron pipe size								
Regular				Iron pipe size	Extra heavy			
Diameter, in.		Weight in pounds per foot			Diameter, in.		Weight in pounds per foot	
Outside	Inside	Brass	Copper		Outside	Inside	Brass	Copper
.405	.281	.246	.259	1/8"	.405	.205	.353	.371
.540	.375	.437	.459	1/4"	.540	.294	.593	.624
.675	.494	.612	.644	3/8"	.675	.421	.805	.847
.840	.625	.911	.958	1/2"	.840	.542	1.191	1.253
1.050	.822	1.235	1.298	3/4"	1.050	.736	1.622	1.706
1.315	1.062	1.740	1.829	1"	1.315	.951	2.386	2.509
1.660	1.368	2.557	2.689	1¼"	1.660	1.272	3.291	3.460
1.900	1.600	3.037	3.193	1½"	1.900	1.494	3.986	4.191
2.375	2.062	4.017	4.224	2"	2.375	1.933	5.508	5.791
2.875	2.500	5.830	6.130	2½"	2.875	2.315	8.407	8.839
3.500	3.062	8.314	8.741	3"	3.500	2.892	11.24	11.82
4.000	3.500	10.85	11.41	3½"	4.000	3.358	13.66	14.37
4.500	4.000	12.29	12.93	4"	4.500	3.818	16.41	17.25
5.000	4.500	13.74	14.44	4½"	5.000	4.250	20.07	21.10
5.563	5.062	15.40	16.19	5"	5.563	4.813	22.51	23.67
6.625	6.125	18.44	19.39	6"	6.625	5.750	31.32	32.93
7.625	7.062	23.92	25.15	7"	7.625	6.625	41.22	43.34
8.625	8.000	30.05	31.60	8"	8.625	7.625	47.00	49.92
9.625	8.937	36.94	38.84	9"				
10.750	10.019	43.91	46.17	10"				
Plumber's size								
.654	.521	.452	.475	5/8"				
.768	.631	.551	.583	3/4"				
.875	.728	.682	.717	7/8"				
1.000	.836	.871	.916	1"				
1.245	1.060	1.233	1.297	1¼"				
1.508	1.311	1.606	1.689	1½"				
1.756	1.564	1.844	1.939	1¾"				
2.007	1.815	2.123	2.232	2"				

* American Brass Co.

Table 28-2. Dimensions of Standard Wrought-Iron Pipe*

Black and galvanized for temperatures up to 450 deg.

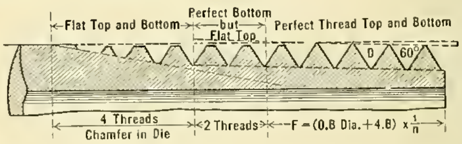
1¼-In. and smaller proved to 300 lb. per sq. in by hydraulic pressure
1½-In. and larger proved to 500 lb. per sq. in. by hydraulic pressure

Nominal diameter	Actual outside diameter	Actual inside diameter	Inside circumference	Outside circumference	Length of pipe per sq. ft. of inside surface	Length of pipe per sq. ft. of outside surface	Inside area	Outside area	Length of pipe containing one cubic foot	Weight per ft.
In.	In.	In.	In.	In.	Ft.	Ft.	In.	In.	Ft.	Lb.
⅛	0.405	0.270	0.848	1.272	14.15	9.44	0.0572	0.129	2500.	0.243
¼	0.54	0.364	1.144	1.696	10.50	7.075	0.1041	0.229	1385.	0.422
⅜	0.675	0.494	1.552	2.121	7.67	5.657	0.1916	0.358	751.5	0.561
½	0.84	0.623	1.957	2.652	6.13	4.502	0.3048	0.554	472.4	0.845
¾	1.05	0.824	2.589	3.299	4.635	3.637	0.5333	0.866	270.	1.126
1	1.315	1.048	3.292	4.134	3.679	2.903	0.8627	1.357	166.9	1.670
1¼	1.66	1.380	4.335	5.215	2.768	2.301	1.496	2.164	96.25	2.258
1½	1.90	1.611	5.061	5.969	2.371	2.01	2.038	2.835	70.65	2.694
2	2.375	2.067	6.494	7.461	1.848	1.611	3.355	4.430	42.36	3.600
2½	2.875	2.468	7.754	9.032	1.547	1.328	4.783	6.491	30.11	5.773
3	3.50	3.067	9.636	10.996	1.245	1.091	7.388	9.621	19.49	7.547
3½	4.00	3.548	11.146	12.566	1.077	0.955	9.887	12.566	14.56	9.055
4	4.50	4.026	12.648	14.137	0.949	0.849	12.730	15.904	11.31	10.66
4½	5.00	4.508	14.153	15.708	0.848	0.765	15.939	19.635	9.03	12.34
5	5.563	5.045	15.849	17.475	0.757	0.629	19.990	24.299	7.20	14.50
6	6.625	6.065	19.054	20.813	0.63	0.577	28.889	34.471	4.98	18.767
7	7.625	7.023	22.063	23.954	0.544	0.505	38.737	45.663	3.72	23.27
8	8.625	7.982	25.076	27.096	0.478	0.444	50.039	58.426	2.88	28.177
9	9.625	9.001	28.277	30.433	0.425	0.394	63.633	73.715	2.26	33.70
10	10.75	10.019	31.475	33.772	0.381	0.355	78.838	90.762	1.80	40.06
11	12.00	11.25	35.343	37.699	0.340	0.318	98.942	113.097	1.455	45.95
12	12.75	12.000	38.264	40.840	0.313	0.293	116.535	132.732	1.235	48.98
14	14.00	13.25	41.268	43.982	0.290	0.273	131.582	153.938	1.069	53.92
15	15.00	14.25	44.271	47.124	0.271	0.254	155.968	176.715	.923	57.89
16	16.00	15.25	47.271	50.265	0.254	0.238	177.867	201.062	.809	61.77
18	18.00	17.25	53.281	56.548	0.225	0.212	225.907	254.469	.638	69.66
20	20.00	19.25	59.288	62.832	0.202	0.191	279.720	314.160	.515	77.57

* Walworth Manufacturing Company

Table 28-3. Standard Pipe Threads (Briggs Formula)

Taper of pipe end = ¾-in. per ft. = $\frac{1}{64}$-in. per in.
Depth of thread (D) = 0.8 x no. of threads per in



Nominal inside diam. of pipe, in.	No. of threads per in.	Dia. at end of pipe, in.	Diam. at bottom of thread, in.	Total distance pipe screws into fitting, in.	Nominal inside diam. of pipe, in.	No. of threads per in.	Diam. at end of pipe, in.	Diam. at bottom of thread, in.	Total distance pipe screws into fitting, in.
⅛	27	0.393	0.331	0.19	3	8	3.441	3.241	0.95
¼	18	0.522	0.433	0.29	3½	8	3.938	3.738	1.00
⅜	18	0.656	0.568	0.30	4	8	4.434	4.234	1.05
½	14	0.815	0.701	0.39	4½	8	4.931	4.731	1.10
¾	14	1.025	0.911	0.40	5	8	5.490	5.290	1.16
1	11½	1.283	1.144	0.51	6	8	6.546	6.346	1.26
1¼	11½	1.626	1.488	0.54	7	8	7.540	7.340	1.36
1½	11½	1.866	1.728	0.55	8	8	8.534	8.334	1.46
2	11½	2.339	2.201	0.58	9	8	9.527	9.327	1.57
2½	8	2.819	2.619	0.89	10	8	10.645	10.445	1.68

Table 28-4. Dimensions of Black and Galvanized Wrought-Iron Pipe

Size	Extra strong				Double extra strong			
	Diameters		Thickness	Weight per foot Plain ends	Diameters		Thickness	Weight per foot Plain ends
	External	Internal			External	Internal		
1/8	.405	.215	.095	.314				
1/4	.540	.302	.119	.535				
3/8	.675	.423	.126	.738				
1/2	.840	.546	.147	1.087	.840	.252	.294	1.714
3/4	1.050	.742	.154	1.473	1.050	.434	.308	2.440
1	1.315	.957	.179	2.171	1.315	.599	.358	3.659
1 1/4	1.660	1.278	.191	2.996	1.660	.896	.382	5.214
1 1/2	1.900	1.500	.200	3.631	1.900	1.100	.400	6.408
2	2.375	1.939	.218	5.022	2.375	1.503	.436	9.029
2 1/2	2.875	2.323	.276	7.661	2.875	1.771	.552	13.695
3	3.500	2.900	.300	10.252	3.500	2.300	.600	18.583
3 1/2	4.000	3.364	.318	12.505	4.000	2.728	.636	22.850
4	4.500	3.826	.337	14.983	4.500	3.152	.674	27.541
4 1/2	5.000	4.290	.355	17.611	5.000	3.580	.710	32.530
5	5.563	4.813	.375	20.778	5.563	4.063	.750	38.552
6	6.625	5.761	.432	28.573	6.625	4.897	.864	53.160
7	7.625	6.625	.500	38.018	7.625	5.875	.875	63.079
8	8.625	7.625	.500	43.388	8.625	6.875	.875	72.424
9	9.625	8.625	.500	48.728				
10	10.750	9.750	.500	54.735				
11	11.750	10.750	.500	60.075				
12	12.750	11.750	.500	65.415				
13	14.000	13.000	.500	72.091				
14	15.000	14.000	.500	77.431				
15	16.000	15.000	.500	82.771				

Table 28-5. Dimensions of Standard Boiler Tubes*

Diameter		Nominal thickness	Nearest no. B. Wire Gauge	Circumference		Transverse area			Length of tube per square foot of		Nominal weight per foot
External	Internal			External	Internal	External	Internal	Metal	external surface	internal surface	
Inches	Inches	Inches		Inches	Inches	Square inches	Square inches	Square inches	Feet	Feet	Pounds
1 3/4	1.560	.095	13	5.498	4.901	2.405	1.911	.494	2.182	2.448	1.679
2	1.810	.095	13	6.283	5.686	3.142	2.573	.569	1.909	2.110	1.932
2 1/4	2.060	.095	13	7.069	6.472	3.976	3.333	.643	1.697	1.854	2.186
2 1/2	2.282	.109	12	7.854	7.169	4.909	4.090	.819	1.527	1.674	2.783
2 3/4	2.532	.109	12	8.639	7.955	5.940	5.036	.904	1.388	1.508	3.074
3	2.782	.109	12	9.425	8.740	7.069	6.079	.990	1.273	1.373	3.365
3 1/4	3.010	.120	11	10.210	9.456	8.296	7.116	1.180	1.175	1.269	4.011
3 1/2	3.260	.120	11	10.996	10.242	9.621	8.347	1.274	1.091	1.171	4.331
3 3/4	3.510	.120	11	11.781	11.027	11.045	9.677	1.368	1.018	1.088	4.652
4	3.732	.134	10	12.566	11.724	12.566	10.939	1.627	.954	1.023	5.532
4 1/2	4.232	.134	10	14.137	13.295	15.904	14.066	1.838	.848	.902	6.248
5	4.704	.148	9	15.708	14.778	19.635	17.379	2.256	.763	.812	7.669

* Crane Co.

Table 28-6. Surface Factors for Pipes

Size of pipe	Factors for reducing lineal ft. to sq. ft.	Factors for reducing sq. ft. to lineal ft.	Size of pipe	Factors for reducing lineal ft. to sq. ft.	Factor for reducing sq. ft. to lineal ft.	Size of pipe	Factors for reducing lineal ft. to sq. ft.	Factors for reducing sq. ft. to lineal ft.
3/4	.27	3.64	3	.92	1.09	7	2.00	.50
1	.33	2.90	3 1/2	1.05	.96	8	2.23	.44
1 1/4	.43	2.30	4	1.19	.85	9	2.50	.40
1 1/2	.50	2.01	4 1/2	1.31	.76	10	2.85	.36
2	.62	1.61	5	1.61	.63	12	3.33	.30
2 1/2	.75	1.33	6	1.75	.58			

Table 28-7. Expansion of Wrought-Iron Pipe on the Application of Heat*

Temp. air when pipe is fitted	Increase in length in inches per foot when heated to							
Deg. fahr.	160	180	200	212	220	228	240	274
0	.0128	.0144	.016	.017	.0176	.0182	.0192	.0219
32	.0102	.0118	.0134	.0144	.015	.0157	.0166	.0194
50	.0088	.0104	.012	.013	.0136	.0142	.0152	.0179
70	.0072	.0088	.0104	.0114	.012	.0126	.0136	.0163

Coefficient:— .0000067 per deg. fahr.

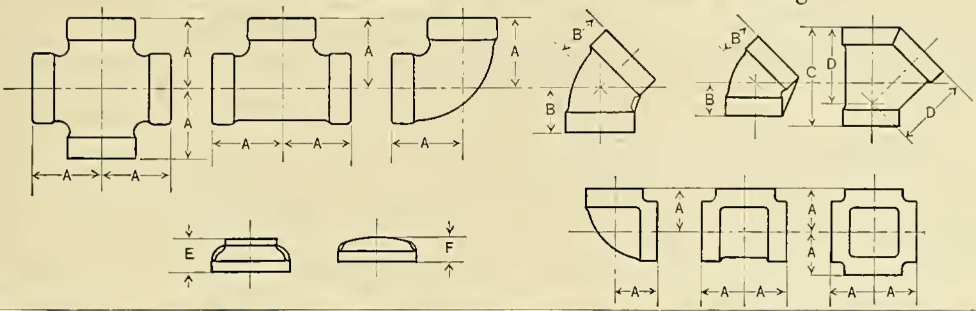
*Holland Heating Manual

Table 28-8. Heat Units Per Pound and Weight Per Cubic Foot of Water Between 32 Deg. Fahr. and 340 Deg. Fahr.†

Temperature, degrees fahr.	Heat units per pound	Weight per cubic foot	Temperature, degrees fahr.	Heat units per pound	Weight per cubic foot	Temperature, degrees fahr.	Heat units per pound	Weight per cubic foot	Temperature, degrees fahr.	Heat units per pound	Weight per cubic foot	Temperature, degrees fahr.	Heat units per pound	Weight per cubic foot	Temperature, degrees fahr.	Heat units per pound	Weight per cubic foot
32	0.00	62.42	70	38.06	62.30	108	75.95	61.90	146	113.86	61.27	184	151.89	60.49	222	190.1	59.58
33	1.01	62.42	71	39.06	62.30	109	76.91	61.88	147	114.86	61.25	185	152.89	60.47	223	191.1	59.55
34	2.01	62.42	72	40.05	62.29	110	77.91	61.86	148	115.86	61.24	186	153.89	60.45	224	192.1	59.53
35	3.02	62.43	73	41.05	62.28	111	78.94	61.85	149	116.86	61.22	187	154.90	60.42	225	193.1	59.50
36	4.03	62.43	74	42.05	62.27	112	79.93	61.83	150	117.86	61.20	188	155.90	60.40	226	194.1	59.48
37	5.04	62.43	75	43.05	62.26	113	80.93	61.82	151	118.86	61.18	189	156.90	60.38	227	195.2	59.45
38	6.04	62.43	76	44.04	62.26	114	81.93	61.80	152	119.86	61.16	190	157.91	60.36	228	196.2	59.42
39	7.05	62.43	77	45.04	62.25	115	82.92	61.79	153	120.86	61.14	191	158.91	60.33	229	197.2	59.40
40	8.05	62.43	78	46.04	62.24	116	83.92	61.77	154	121.86	61.12	192	159.91	60.31	230	198.2	59.37
41	9.05	62.43	79	47.04	62.23	117	84.92	61.75	155	122.86	61.10	193	160.91	60.29	231	199.2	59.34
42	10.06	62.43	80	48.03	62.22	118	85.92	61.74	156	123.86	61.08	194	161.92	60.27	232	200.2	59.32
43	11.06	62.43	81	49.03	62.21	119	86.91	61.72	157	124.86	61.06	195	162.92	60.24	233	201.2	59.29
44	12.06	62.43	82	50.03	62.20	120	87.91	61.71	158	125.86	61.04	196	163.92	60.22	234	202.2	59.27
45	13.07	62.42	83	51.02	62.19	121	88.91	61.69	159	126.86	61.02	197	164.93	60.19	235	203.2	59.24
46	14.07	62.42	84	52.02	62.18	122	89.91	61.68	160	127.86	61.00	198	165.93	60.17	236	204.2	59.21
47	15.07	62.42	85	53.02	62.17	123	90.90	61.66	161	128.86	60.98	199	166.94	60.15	237	205.3	59.19
48	16.07	62.42	86	54.01	62.16	124	91.90	61.65	162	129.86	60.96	200	167.94	60.12	238	206.3	59.16
49	17.08	62.42	87	55.01	62.15	125	92.90	61.63	163	130.86	60.94	201	168.94	60.10	239	207.3	59.14
50	18.08	62.42	88	56.01	62.14	126	93.90	61.61	164	131.86	60.92	202	169.95	60.07	240	208.3	59.11
51	19.08	62.41	89	57.00	62.13	127	94.89	61.59	165	132.86	60.90	203	170.95	60.05	241	209.3	59.08
52	20.08	62.41	90	58.00	62.12	128	95.89	61.58	166	133.86	60.88	204	171.96	60.02	242	210.3	59.05
53	21.08	62.41	91	59.00	62.11	129	96.89	61.56	167	134.86	60.86	205	172.96	60.00	243	211.4	59.03
54	22.08	62.40	92	60.00	62.09	130	97.89	61.55	168	135.86	60.84	206	173.97	59.98	244	212.4	59.00
55	23.08	62.40	93	60.99	62.08	131	98.89	61.53	169	136.86	60.82	207	174.97	59.95	245	213.4	58.97
56	24.08	62.39	94	61.99	62.07	132	99.88	61.52	170	137.87	60.80	208	175.98	59.93	246	214.4	58.94
57	25.08	62.39	95	62.99	62.06	133	100.88	61.50	171	138.87	60.78	209	176.98	59.90	247	215.4	58.91
58	26.08	62.38	96	63.98	62.05	134	101.88	61.49	172	139.87	60.76	210	177.99	59.88	248	216.4	58.89
59	27.08	62.37	97	64.98	62.04	135	102.88	61.47	173	140.87	60.73	211	178.99	59.85	249	217.4	58.86
60	28.08	62.37	98	65.98	62.03	136	103.88	61.45	174	141.87	60.71	212	180.00	59.83	250	218.5	58.83
61	29.08	62.36	99	66.97	62.02	137	104.87	61.43	175	142.87	60.69	213	181.0	59.80	260	228.6	58.55
62	30.08	62.36	100	67.97	62.00	138	105.87	61.41	176	143.87	60.67	214	182.0	59.78	270	238.8	58.26
63	31.07	62.35	101	68.97	61.99	139	106.87	61.40	177	144.88	60.65	215	183.0	59.75	280	249.0	57.96
64	32.07	62.35	102	69.96	61.98	140	107.87	61.38	178	145.88	60.62	216	184.0	59.73	290	259.3	57.65
65	33.07	62.34	103	70.96	61.97	141	108.87	61.36	179	146.88	60.60	217	185.0	59.70	300	269.6	57.33
66	34.07	62.33	104	71.96	61.95	142	109.87	61.34	180	147.88	60.58	218	186.1	59.68	310	279.9	57.00
67	35.07	62.33	105	72.95	61.94	143	110.87	61.33	181	148.88	60.56	219	187.1	59.65	320	290.2	56.66
68	36.07	62.32	106	73.95	61.93	144	111.87	61.31	182	149.89	60.53	220	188.1	59.63	330	300.6	56.30
69	37.06	62.31	107	74.95	61.91	145	112.86	61.29	183	150.89	60.51	221	189.1	59.60	340	311.0	55.94

† *Steam*, Babcock & Wilcox Co.

Table 28-9. Dimensions of Cast-Iron Screwed Fittings*

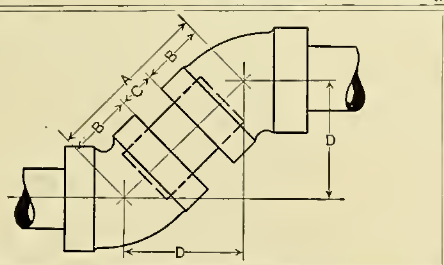


Size, inches	Standard A Inches	Standard B Inches	Extra heavy A Inches	Extra heavy B Inches	Standard and extra heavy C Inches	Standard and extra heavy D Inches	Standard and extra heavy E Inches	Standard and extra heavy F Inches
¼	13/16	¾	....	...	....	....	...	...
⅜	15/16	13/16	....	...	....	....	...	...
½	1⅛	⅞	....	...	2½	1⅞	...	...
¾	1 5/16	1	....	...	3	2¼	...	...
1	1 7/16	1⅛	....	...	3½	2¾	...	...
1¼	1¾	1 5/16	2	1⅜	4¼	3¼	...	...
1½	1 15/16	1 7/16	2¼	1½	4⅞	3 13/16	...	...
2	2¼	1 11/16	2 9/16	1⅝	5¾	4½	...	...
2½	2 11/16	1 15/16	3	1 15/16	6¼	5 3/16	...	...
3	3⅛	2 3/16	3½	2¼	7⅞	6⅛	2 15/16	...
3½	3 7/16	2⅜	4⅛	2½	8⅞	6⅞	3⅛	...
4	3¾	2⅝	4 11/16	2 9/16	9¾	7⅝	3⅜	2 1/16
4½	4 1/16	2 13/16	5⅛	2¾	11⅝	9¼	3⅝	2 3/16
5	4 7/16	3 1/16	5½	3	11⅝	9¼	3⅞	2⅜
6	5⅛	3 7/16	6⅛	3 5/16	13 7/16	10¾	4⅜	2⅝
7	5 13/16	3⅞	7¼	3¾	15¼	12¼	4 13/16	2⅞
8	6½	4¼	8⅛	4	16 15/16	13⅝	5¼	3⅛
9	7 3/16	4 11/16	9⅛	4¾	20 11/16	16¾	5 11/16	3⅜
10	7⅞	5 3/16	11⅜	4⅞	20 11/16	16¾	6 3/16	3⅝
12	9¼	6	13⅜	5½	24⅛	19⅝	7⅛	4¼

NOTE—The above dimensions are subject to a slight variation

* Crane Co.

Table 28-10. 45-Degree Offset Connections



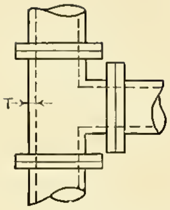
Pipe size	Centre to centre A	Centre to face B	Face to face of 45's C	Offset D
½	2¼	⅞	½	1 19/32
¾	2½	1	½	1 25/32
1	2¾	1⅛	½	1 31/32
1¼	3⅛	1 5/16	½	2 7/32
1½	3⅜	1 7/16	½	2 13/32
2	3⅞	1 11/16	½	2¾
2½	4⅝	1 15/16	¾	3 9/32
3	5⅛	2 3/16	¾	3⅝
3½	5½	2⅜	¾	3 29/32
4	6	2⅝	¾	4¼
4½	6⅜	2 13/16	¾	4½
5	7⅛	3 1/16	1	5 1/16
6	7⅞	3 7/16	1	5 19/32
7	8¾	3⅞	1	6 7/32
8	10	4½	1	7 1/16

NOTE: The Offset D is equal to the distance A ÷ 1.414

Table 28-11. Rules for Standard Weight Flanged Fittings

American 1915 Standard, 125-lb. working pressure

Shell thickness in inches



Size fitting, inches	Shell thickness	Size fitting, inches	Shell thickness	Size fitting, inches	Shell thickness
2	7/16	5	½	12	⅞
2½	7/16	6	9/16	14	15/16
3	½	7	⅝	15	15/16
3½	½	8	11/16	16	15/16
4	½	9	¾	18	1
4½	½	10	13/16	20	1 1/16

1. Standard reducing elbows carry same dimensions center-to-face as regular elbows of largest straight size.

2. Standard tees, crosses and laterals, reducing on run only, carry same dimensions face-to-face as largest straight size.

3. Where long-turn fittings are specified, it has reference only to elbows which are made in two center-to-face dimensions and to be known as elbows and long-turn elbows, the latter being used only when so specified.

4. All standard weight fittings must be guaranteed for 125-lb. working pressure, and each must have mark cast on indicating maker and guaranteed working steam pressure.

5. Standard weight fittings and flanges to be plain faced, and bolt holes to be ⅛ in. larger in diameter than bolts; bolt holes to straddle center lines.

6. Size of all fittings scheduled indicates inside diameter of ports.

7. Square head bolts with hexagonal nuts are generally recommended for use.

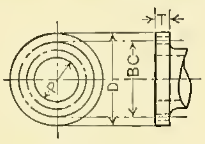
8. Double-branch elbows, side-outlet elbows and side-outlet tees, whether straight or reducing, carry same dimensions center-to-face and face-to-face as regular tees and elbows.

9. Bull-head tees or tees increasing on outlet, will have same center-to-face and face-to-face dimensions as a straight fitting of the size of the outlet.

10. Tees, crosses and laterals 16-in. and smaller, reducing on the outlet, use the same dimensions as straight sizes of the larger port. (*Continued on next page*)

Table 28-12. Standard Flanges and Bolts

1915 Standard, 125-lb. working pressure



Pipe	Flange		Bolts		Bolt Holes	
Size P	Diam. D	Thickness T	No.	Size Diam.	Bolt circle B. C.	Size Diam.
1	4	7/16	4	7/16	3	9/16
1¼	4½	½	4	7/16	3⅜	9/16
1½	5	9/16	4	½	3⅞	⅝
2	6	⅝	4	⅝	4¾	¾
2½	7	11/16	4	⅝	5½	¾
3	7½	¾	4	⅝	6	¾
3½	8½	13/16	4	⅝	7	¾
4	9	15/16	8	⅝	7½	¾
4½	9¼	15/16	8	¾	7¾	⅞
5	10	15/16	8	¾	8½	⅞
6	11	1	8	¾	9½	⅞
7	12½	1 1/16	8	¾	10¾	⅞
8	13½	1⅛	8	¾	11¾	⅞
9	15	1⅛	12	¾	13¼	⅞
10	16	1 3/16	12	⅞	14¼	1
12	19	1¼	12	⅞	17	1
14	21	1⅜	12	1	18¾	1⅛
15	22¼	1⅜	16	1	20	1⅛
16	23½	1 7/16	16	1	21¼	1⅛
18	25	1 9/16	16	1⅛	22¾	1¼
20	27½	1 11/16	20	1⅛	25	1¼
22	29½	1 13/16	20	1¼	27¼	1⅜
24	32	1⅞	20	1¼	29½	1⅜
26	34¼	2	24	1¼	31¾	1⅜
28	36½	2 1/16	28	1¼	34	1⅜
30	38¾	2⅛	28	1⅜	36	1½
32	41¾	2¼	28	1½	38½	1⅝
34	43¾	2 5/16	32	1½	40½	1⅝
36	46	2⅜	32	1½	42¾	1⅝
38	48¾	2⅜	32	1⅝	45¼	1¾
40	50¾	2½	36	1⅝	47¼	1¾

Sizes 18-in. and larger, reducing on the outlet, are made in two lengths, depending on the size of the outlet, as given in the table of dimensions.

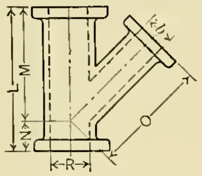
11. For fittings reducing on the run only, a long-body pattern will be used. Y's are special and made to suit connections. Double-branch elbows are not made reducing on the run.

12. Steel flanges, fittings and valves are recommended for superheated steam.

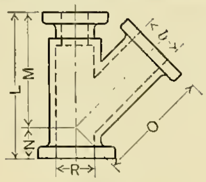
13. If flanged fittings for lower working pressure than 125 lb. are made, they shall conform in all dimensions, except thickness of shell, to this standard and shall have the guaranteed working pressure cast on each fitting. Flanges for these fittings must be standard dimensions.

Table 28-13. Standard Flanged Reducing Laterals

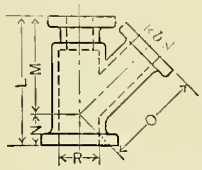
1915 Standard, 125-lb. Working Pressure



Reducing lateral



Reducing-on-run lateral

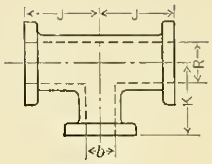


Reducing-on-run and branch lateral

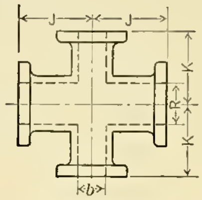
Size		Dimensions, inches				Flanges	
Run-R	Branch b	L	M	N	O	Diam.	Thickness
1	—	—	—	—	—	4	7/16
1¼	1¼ or less	8	6¼	1¾	6¼	4½	½
1½	1½ " "	9	7	2	7	5	9/16
2	2 " "	10½	8	2½	8	6	⅝
2½	2½ " "	12	9½	2½	9½	7	11/16
3	3 " "	13	10	3	10	7½	¾
3½	3½ " "	14½	11½	3	11½	8½	13/16
4	4 " "	15	12	3	12	9	15/16
4½	4½ " "	15½	12½	3	12½	9¼	15/16
5	5 " "	17	13½	3½	13½	10	15/16
6	6 " "	18	14½	3½	14½	11	1
7	7 " "	20½	16½	4	16½	12½	1 1/16
8	8 " "	22	17½	4½	17½	13½	1⅛
9	9 " "	24	19½	4½	19½	15	1⅛
10	10 " "	25½	20½	5	20½	16	1 3/16
12	12 " "	30	24½	5½	24½	19	1¼
14	14 " "	33	27	6	27	21	1⅜
15	15 " "	34½	28½	6	28½	22¼	1⅜
16	16 " "	36½	30	6½	30	23½	1 7/16
18	9 " "	26	25	1	27½	25	1 9/16
18	18 to 10 inc.	39	32	7	32	25	1 9/16
20	10 and less	28	27	1	29½	27½	1 11/16
20	20 to 12 inc.	43	35	8	35	27½	1 11/16
22	10 and less	29	28½	½	31½	29½	1 13/16
22	22 to 12 inc.	46	37½	8½	37½	29½	1 13/16
24	12 and less	32	31½	½	34½	32	1⅞
24	24 to 14 inc.	49½	40½	9	40½	32	1⅞
26	12 and less	35	35	0	38	34¼	2
26	26 to 14 inc.	53	44	9	44	34¼	2
28	14 and less	37	37	0	40	36½	2 1/16
28	28 to 15 inc.	56	46½	9½	46½	36½	2 1/16
30	15 and less	39	39	0	42	38¾	2⅛
30	30 to 16 inc.	59	49	10	49	38¾	2⅛

Table 28-14. Standard Flanged Bull-Head Reducing Tees and Crosses

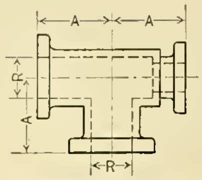
1915 Standard, 125-lb. Working Pressure



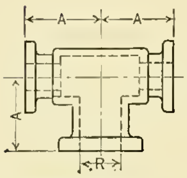
Reducing tee



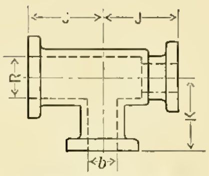
Reducing cross



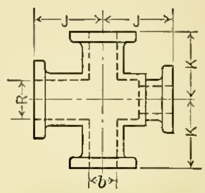
Reducing-on-run tee



Bull-head tee



Reducing-on-run and Branch tee cross



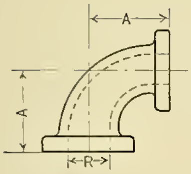
Reducing-on-run branch

Size		Dimensions, inches					Flanges	
Run-R	Branch b	A & J	K	Branch b	J	K	Diam.	Thickness
1	—	—	—				4	7/16
1¼	1 or less	3¾	3¾				4½	½
1½	1¼ " "	4	4				5	9/16
2	1½ " "	4½	4½				6	⅝
2½	2 " "	5	5				7	11/16
3	2½ " "	5½	5½				7½	¾
3½	3 " "	6	6				8½	13/16
4	3½ " "	6½	6½				9	15/16
4½	4 " "	7	7				9¼	15/16
5	4½ " "	7½	7½				10	15/16
6	5 " "	8	8				11	1
7	6 " "	8½	8½				12½	1 1/16
8	7 " "	9	9				13½	1⅛
9	8 " "	10	10				15	1⅛
10	9 " "	11	11				16	1 3/16
12	10 " "	12	12				19	1¼
14	12 " "	14	14				21	1⅜
15	14 " "	14½	14½				22¼	1⅜
16	15 " "	15	15				23½	1 7/16
18	18 to 14 inc.	16½	16½	12 or less	13	15½	25	1 9/16
20	20 to 15 inc.	18	18	14 " "	14	17	27½	1 11/16
22	22 to 16 inc.	20	20	15 " "	14	18	29½	1 13/16
24	24 to 18 inc.	22	22	16 " "	15	19	32	1⅞
26	26 to 20 inc.	23	23	18 " "	16	20	34¼	2
28	28 to 20 inc.	24	24	18 " "	16	21	36½	2 1/16
30	30 to 22 inc.	25	25	20 " "	18	23	38¾	2⅛
32	32 to 22 inc.	26	26	20 " "	18	24	41¾	2¼
34	34 to 24 inc.	27	27	22 " "	19	25	43¾	2 5/16
36	36 to 26 inc.	28	28	24 " "	20	26	46	2⅜
38	38 to 26 inc.	29	29	24 " "	20	28	48¾	2⅜
40	40 to 28 inc.	30	30	26 " "	22	29	50¾	2½

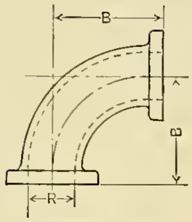
Note—A reduction in size *on the run* does not affect the dimensions but branch outlets of small size such as are listed below will reduce the dimensions of fittings 18 in. or over in size

Table 28-15. Standard Flanged Elbows, Crosses, Laterals and Reducers

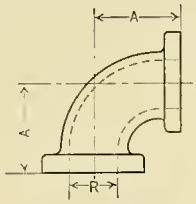
1915 Standard, 125-lb. Working Pressure



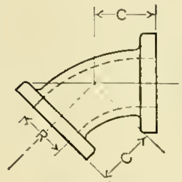
Elbow



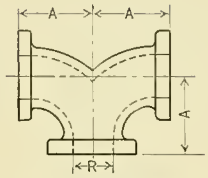
Long-turn elbow



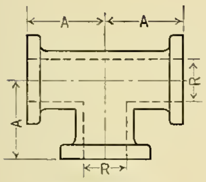
Reducing elbow



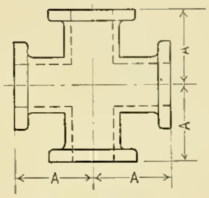
45-deg. elbow



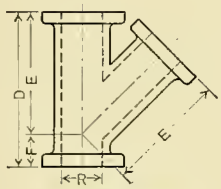
Double-branch elbow



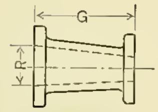
Straight tee



Straight cross



Straight lateral



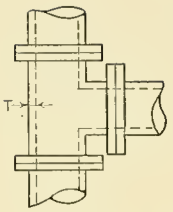
Reducer

Size Run-R	Dimensions, inches A	B	C	D	E	G	Flange Diam.	Flange Thickness
1	3½	5	1¾	7½	5¾	—	4	$\frac{7}{16}$
1¼	3¾	5½	2	8	6¼	—	4½	½
1½	4	6	2¼	9	7	—	5	$\frac{9}{16}$
2	4½	6½	2½	10½	8	—	6	⅝
2½	5	7	3	12	9½	—	7	$\frac{11}{16}$
3	5½	7¾	3	13	10	6	7½	¾
3½	6	8½	3½	14½	11½	6½	8½	$\frac{13}{16}$
4	6½	9	4	15	12	7	9	$\frac{15}{16}$
4½	7	9½	4	15½	12½	7½	9¼	$\frac{15}{16}$
5	7½	10¼	4½	17	13½	8	10	$\frac{15}{16}$
6	8	11½	5	18	14½	9	11	1
7	8½	12¾	5½	20½	16½	10	12½	1$\frac{1}{16}$
8	9	14	5½	22	17½	11	13½	1⅛
9	10	15¼	6	24	19½	11½	15	1⅛
10	11	16½	6½	25½	20½	12	16	1$\frac{3}{16}$
12	12	19	7½	30	24½	14	19	1¼
14	14	21½	7½	33	27	16	21	1⅜
15	14½	22¾	8	34½	28½	17	22¼	1⅜
16	15	24	8	36½	30	18	23½	1$\frac{7}{16}$
18	16½	26½	8½	39	32	19	25	1$\frac{9}{16}$
20	18	29	9½	43	35	20	27½	1$\frac{11}{16}$
22	20	31½	10	46	37½	22	29½	1$\frac{13}{16}$
24	22	34	11	49½	40½	24	32	1⅞
26	23	36½	13	53	44	26	34¼	2
28	24	39	14	56	46	28	36½	2$\frac{1}{16}$
30	25	41½	15	59	49	30	38¾	2⅛
32	26	44	16	—	—	32	41¾	2¼
34	27	46½	17	—	—	34	43¾	2$\frac{5}{16}$
36	28	49	18	—	—	36	46	2⅜
38	29	51½	19	—	—	38	48¾	2⅜
40	30	54	20	—	—	40	50¾	2½

Table 28-16. Rules for Extra-Heavy Flanged Fittings

American 1915 Standard 250-lb. Working Pressure

Shell thickness in inches



Size fitting, inches	Shell thickness	Size fitting, inches	Shell thickness	Size fitting, inches	Shell thickness
2	5/8	5	3/4	12	1 1/8
2 1/2	5/8	6	13/16	14	1 3/16
3	5/8	7	7/8	15	1 1/4
3 1/2	5/8	8	15/16	16	1 5/16
4	5/8	9	1	18	1 7/16
4 1/2	11/16	10	1 1/16	20	1 1/2

1. Extra heavy reducing elbows carry same dimensions center-to-face as regular elbows of largest straight size.

2. Extra heavy tees, crosses and laterals, reducing on run only, carry same dimensions face-to-face as largest straight size.

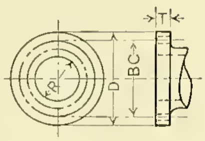
3. Where long-turn fittings are specified, it has reference only to elbows which are made in two center-to-face dimensions and to be known as elbows and long-turn elbows, the latter being used only when so specified.

4. Extra heavy fittings must be guaranteed for 250-lb. working pressure, and each fitting must have some mark cast on it indicating the maker and guaranteed working steam pressure.

5. All extra heavy fittings and flanges to have a raised surface $\frac{1}{16}$ in. high inside of bolt holes for gaskets. Thickness of flanges and center-to-face dimensions of fittings include this raised surface. Bolt holes to be $\frac{1}{8}$ in. larger in diameter than bolts. Bolt holes to straddle center lines. (*Continued on next page.*)

Table 28-17. Extra-Heavy Pipe Flanges and Bolts

1915 Standard, 250-lb. Working Pressure



Pipe Size P	Flange Diam. D	Flange Thickness T	Bolts No.	Bolts Size	Bolt holes: Bolt circle B. C.	Bolt holes: Bolt hole
1	4 1/2	11/16	4	1/2	3 1/4	5/8
1 1/4	5	3/4	4	1/2	3 3/4	5/8
1 1/2	6	13/16	4	5/8	4 1/2	3/4
2	6 1/2	7/8	4	5/8	5	3/4
2 1/2	7 1/2	1	4	3/4	5 7/8	7/8
3	8 1/4	1 1/8	8	3/4	6 5/8	7/8
3 1/2	9	1 3/16	8	3/4	7 1/4	7/8
4	10	1 1/4	8	3/4	7 7/8	7/8
4 1/2	10 1/2	1 5/16	8	3/4	8 1/2	7/8
5	11	1 3/8	8	3/4	9 1/4	7/8
6	12 1/2	1 7/16	12	3/4	10 5/8	7/8
7	14	1 1/2	12	7/8	11 7/8	1
8	15	1 5/8	12	7/8	13	1
9	16 1/4	1 3/4	12	1	14	1 1/8
10	17 1/2	1 7/8	16	1	15 1/4	1 1/8
12	20 1/2	2	16	1 1/8	17 3/4	1 1/4
14	23	2 1/8	20	1 1/8	20 1/4	1 1/4
15	24 1/2	2 3/16	20	1 1/4	21 1/2	1 3/8
16	25 1/2	2 1/4	20	1 1/4	22 1/2	1 3/8
18	28 1/2	2 3/8	24	1 1/4	24 3/4	1 3/8
20	30 1/2	2 1/2	24	1 3/8	27	1 1/2
22	33	2 5/8	24	1 1/2	29 1/4	1 5/8
24	36	2 3/4	24	1 5/8	32	1 3/4
26	38 1/4	2 13/16	28	1 5/8	34 1/2	1 3/4
28	40 3/4	2 15/16	28	1 5/8	37	1 3/4
30	43	3	28	1 3/4	39 1/4	1 7/8
32	45 1/4	3 1/8	28	1 7/8	41 1/2	2
34	47 1/2	3 1/4	28	1 7/8	43 1/2	2
36	50	3 3/8	32	1 7/8	46	2
38	52 1/4	3 7/16	32	1 7/8	48	2
40	54 1/2	3 9/16	36	1 7/8	50 1/4	2

6. Size of all fittings scheduled indicates inside diameter of ports.

7. Square head bolts with hexagonal nuts are generally recommended for use.

8. Double branch elbows, side outlet elbows and side outlet tees, whether straight or reducing sizes, carry same dimensions center-to-face and face-to-face as regular tees and elbows.

9. Bull-head tees or tees increasing on outlet, will have same center-to-face and face-to-face dimensions as a straight fitting of the size of the outlet.

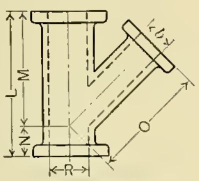
10. Tees, crosses and laterals 16-in. and smaller, reducing on the outlet, use the same dimensions as straight size of the larger port. Sizes 18 in. and larger, reducing on the outlet, are made in two lengths, depending on the size of the outlet as given in the table of dimensions.

11. For fittings reducing on the run only a long body pattern will be used. Y's are special and made to suit connections. Double branch elbows are not made reducing on the run.

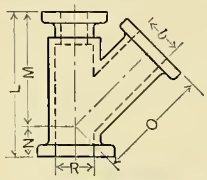
12. Steel flanges, fittings and valves are recommended for superheated steam.

Table 28-18. Extra-Heavy Flanged Reducing Laterals

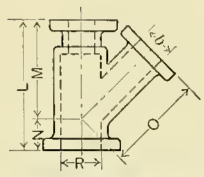
1915 Standard, 250-lb. Working Pressure



Reducing lateral



Reducing-on-run lateral

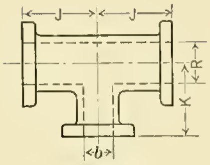


Reducing on-run and Branch lateral

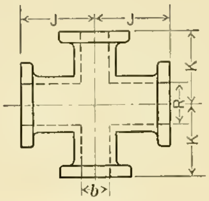
Size		Dimensions, inches				Flanges	
Run-R	Branch b	L	M	N	O	Diam.	Thickness
1	—	—	—	—	—	4½	11/16
1¼	1¼ and less	9½	7¼	2¼	7¼	5	¾
1½	1½ " "	11	8½	2½	8½	6	13/16
2	2 " "	11½	9	2½	9	6½	⅞
2½	2½ " "	13	10½	2½	10½	7½	1
3	3 " "	14	11	3	11	8¼	1⅛
3½	3½ " "	15½	12½	3	12½	9	1 3/16
4	4 " "	16½	13½	3	13½	10	1¼
4½	4½ " "	18	14½	3½	14½	10½	1 5/16
5	5 " "	18½	15	3½	15	11	1⅜
6	6 " "	21½	17½	4	17½	12½	1 7/16
7	7 " "	23½	19	4½	19	14	1½
8	8 " "	25½	20½	5	20½	15	1⅝
9	9 " "	27½	22½	5	22½	16¼	1¾
10	10 " "	29½	24	5½	24	17½	1⅞
12	12 " "	33½	27½	6	27½	20½	2
14	14 " "	37½	31	6½	31	23	2⅛
15	15 " "	39½	33	6½	33	24½	2 3/16
16	16 " "	42	34½	7½	34½	25½	2¼
18	9 " "	34	31	3	32½	28	2⅜
18	16 to 10 inc.	45½	37½	8	37½	28	2⅜
20	10 and less	37	34	3	36	30½	2½
20	18 to 12 inc.	49	40½	8½	40½	30½	2½
22	10 and less	40	37	3	39	33	2⅝
22	20 to 12 inc.	53	43½	9½	43½	33	2⅝
24	12 and less	44	41	3	43	36	2¾
24	22 to 14 inc.	57½	47½	10	47½	36	2¾

Table 28-19. Extra-Heavy Flanged Bull-Head Reducing Tees and Crosses

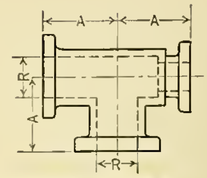
1915 Standard, 250-lb. Working Pressure



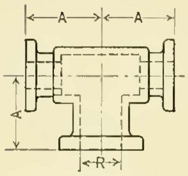
Reducing tee



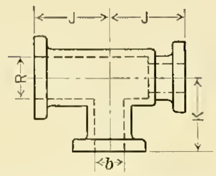
Reducing cross



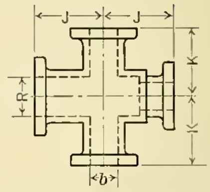
Reducing-on-run tee



Bull-head tee



Reducing-on-run and branch tee



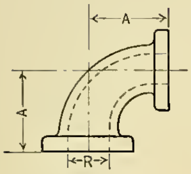
Reducing-on-run and branch cross

Size		Dimensions, inches					Flanges	
Run-R	Branch b	J	K	Branch	J	K	Diam.	Thickness
1	—	—	—				4½	11/16
1¼	1 or less	4¼	4¼				5	¾
1½	1¼ " "	4½	4½				6	13/16
2	1½ " "	5	5				6½	⅞
2½	2 " "	5½	5½				7½	1
3	2½ " "	6	6				8¼	1⅛
3½	3 " "	6½	6½				9	1 3/16
4	3½ " "	7	7				10	1¼
4½	4 " "	7½	7½				10½	1 5/16
5	4½ " "	8	8				11	1⅜
6	5 " "	8½	8½				12½	1 7/16
7	6 " "	9	9				14	1½
8	7 " "	10	10				15	1⅝
9	8 " "	10½	10½				16¼	1¾
10	9 " "	11½	11½				17½	1⅞
12	10 " "	13	13				20½	2
14	12 " "	15	15				23	2⅛
15	14 " "	15½	15½				24½	2 3/16
16	15 " "	16½	16½				25½	2¼
18	18 to 14 inc.	18	18	12 or less	14	17	28	2⅜
20	20 to 15 inc.	19½	19½	14 " "	15½	18½	30½	2½
22	22 to 16 inc.	20½	20½	15 " "	16½	20	33	2⅝
24	24 to 18 inc.	22½	22½	16 " "	17	21½	36	2¾
26	26 to 20 inc.	24	24	18 " "	19	23	38¼	2 13/16
28	28 to 20 inc.	26	26	18 " "	19	24	40¾	2 15/16
30	30 to 22 inc.	27½	27½	20 " "	20½	25½	43	3
32	32 to 22 inc.	29	29	20 " "	20½	26½	45¼	3⅛
34	34 to 24 inc.	30½	30½	22 " "	22	28	47½	3¼
36	36 to 26 inc.	32½	32½	24 " "	23½	29½	50	3⅜
38	38 to 26 inc.	34	34	24 " "	23½	30½	52¼	3 7/16
40	40 to 28 inc.	35½	35½	26 " "	25	31½	54½	3 9/16

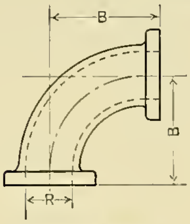
Note—A reduction in size *on the run* does not affect the dimensions but branch outlets of smaller size than those listed below will reduce the dimensions of fittings 18 in. or over in size

Table 28-20. Extra-Heavy Flanged Elbows, Crosses, Laterals and Reducers

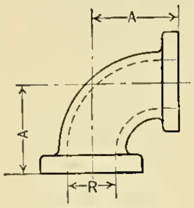
1915 Standard, 250-lb. Working Pressure



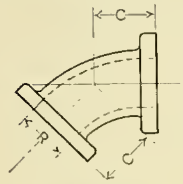
Elbow



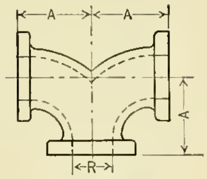
Long-turn elbow



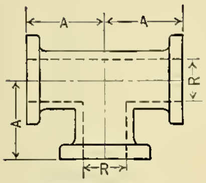
Reducing elbow



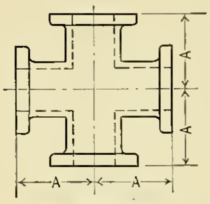
45-deg. elbow



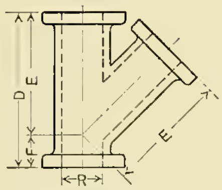
Double-branch elbow



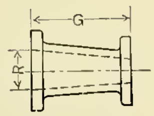
Straight tee



Straight cross



Straight lateral



Reducer

Size	Dimensions, inches						Flange	
Run-R	A	B	C	D	E	G	Diam.	Thickness
1	4	5	2	8½	6½	—	4½	11/16
1¼	4¼	5½	2½	9½	7¼	—	5	¾
1½	4½	6	2¾	11	8½	—	6	13/16
2	5	6½	3	11½	9	—	6½	⅞
2½	5½	7	3½	13	10½	—	7½	1
3	6	7¾	3½	14	11	6	8¼	1⅛
3½	6½	8½	4	15½	12½	6½	9	1 3/16
4	7	9	4½	16½	13½	7	10	1¼
4½	7½	9½	4½	18	14½	7½	10½	1 5/16
5	8	10¼	5	18½	15	8	11	1⅜
6	8½	11½	5½	21½	17½	9	12½	1 7/16
7	9	12¾	6	23½	19	10	14	1½
8	10	14	6	25½	20½	11	15	1⅝
9	10½	15¼	6½	27½	22½	11½	16¼	1¾
10	11½	16½	7	29½	24	12	17½	1⅞
12	13	19	8	33½	27½	14	20½	2
14	15	21½	8½	37½	31	16	23	2⅛
15	15½	22¾	9	39½	33	17	24½	2 3/16
16	16½	24	9½	42	34½	18	25½	2¼
18	18	26½	10	45½	37½	19	28	2⅜
20	19½	29	10½	49	40½	20	30½	2½
22	20½	31½	11	53	43½	22	33	2⅝
24	22½	34	12	57½	47½	24	36	2¾
26	24	36½	13	—	—	26	38¼	2 13/16
28	26	39	14	—	—	28	40¾	2 15/16
30	27½	41½	15	—	—	30	43	3
32	29	44	16	—	—	32	45¼	3⅛
34	30½	46½	17	—	—	34	47½	3¼
36	32½	49	18	—	—	36	50	3⅜
38	34	51½	19	—	—	38	52¼	3 7/16
40	35½	54	20	—	—	40	54½	3 9/16

Table 28-21. Properties of Saturated Steam

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Pressure, lb. absolute	Temperature, deg. fahr.	Specific volume, cu. ft. per lb.	Heat of the liquid, B.t.u.	Latent heat of evap., B.t.u.	Total heat of steam, B.t.u.	Pressure, lb. absolute
1	101.83	333.0	69.8	1034.6	1104.4	1
2	126.15	173.5	94.0	1021.0	1115.0	2
3	141.52	118.5	109.4	1012.3	1121.6	3
4	153.01	90.5	120.9	1005.7	1126.5	4
5	162.28	73.33	130.1	1000.3	1130.5	5
6	170.06	61.89	137.9	995.8	1133.7	6
7	176.85	53.56	144.7	991.8	1136.5	7
8	182.86	47.27	150.8	988.2	1139.0	8
9	188.27	42.36	156.2	985.0	1141.1	9
10	193.22	38.38	161.1	982.0	1143.1	10
11	197.75	35.10	165.7	979.2	1144.9	11
12	201.96	32.36	169.9	976.6	1146.5	12
13	205.87	30.03	173.8	974.2	1148.0	13
14	209.55	28.02	177.5	971.9	1149.4	14
14.7	212.0	26.79	180.0	970.4	1150.4	14.7
15	213.0	26.27	181.0	969.7	1150.7	15
16	216.3	24.79	184.4	967.6	1152.0	16
17	219.4	23.38	187.5	965.6	1153.1	17
18	222.4	22.16	190.5	963.7	1154.2	18
19	225.2	21.07	193.4	961.8	1155.2	19
20	228.0	20.08	196.1	960.0	1156.2	20
22	233.1	18.37	201.3	956.7	1158.0	22
24	237.8	16.93	206.1	953.5	1159.6	24
26	242.2	15.72	210.6	950.6	1161.2	26
28	246.4	14.67	214.8	947.8	1162.6	28
30	250.3	13.74	218.8	945.1	1163.9	30
32	254.1	12.93	222.6	942.5	1165.1	32
34	257.6	12.22	226.2	940.1	1166.3	34
36	261.0	11.58	229.6	937.7	1167.3	36
38	264.2	11.01	232.9	935.5	1168.4	38
40	267.3	10.49	236.1	933.3	1169.4	40
42	270.2	10.02	239.1	931.2	1170.3	42
44	273.1	9.59	242.0	929.2	1171.2	44
46	275.8	9.20	244.8	927.2	1172.0	46
48	278.5	8.84	247.5	925.3	1172.8	48
50	281.0	8.51	250.1	923.5	1173.6	50
52	283.5	8.20	252.6	921.7	1174.3	52
54	285.9	7.91	255.1	919.9	1175.0	54
56	288.2	7.65	257.5	918.2	1175.7	56
58	290.5	7.40	259.8	916.5	1176.4	58
60	292.7	7.17	262.1	914.9	1177.0	60
62	294.9	6.95	264.3	913.3	1177.6	62
64	297.0	6.75	266.4	911.8	1178.2	64
66	299.0	6.56	268.5	910.2	1178.8	66
68	301.0	6.38	270.6	908.7	1179.3	68
70	302.9	6.20	272.6	907.2	1179.8	70
72	304.8	6.04	274.5	905.8	1180.4	72
74	306.7	5.89	276.5	904.4	1180.9	74
76	308.5	5.74	278.3	903.0	1181.4	76
78	310.3	5.60	280.2	901.7	1181.8	78
80	312.0	5.47	282.0	900.3	1182.3	80

Table 28-21. Properties of Saturated Steam—*Continued*

Pressure, lb. absolute	Temperature, deg. fahr.	Specific volume, cu. ft. per lb.	Heat of the liquid, b.t.u.	Latent heat of evap., b.t.u.	Total heat of steam, b.t.u.	Pressure, lb. absolute
82	313.8	5.34	283.8	899.0	1182.8	82
84	315.4	5.22	285.5	897.7	1183.2	84
86	317.1	5.10	287.2	896.4	1183.6	86
88	318.7	5.00	288.9	895.2	1184.0	88
90	320.3	4.89	290.5	893.9	1184.4	90
92	321.8	4.79	292.1	892.7	1184.8	92
94	323.4	4.69	293.7	891.5	1185.2	94
96	324.9	4.60	295.3	890.3	1185.6	96
98	326.4	4.51	296.8	889.2	1186.0	98
100	327.8	4.429	298.3	888.0	1186.3	100
105	331.4	4.230	302.0	885.2	1187.2	105
110	334.8	4.047	305.5	882.5	1188.0	110
115	338.1	3.880	309.0	879.8	1188.8	115
120	341.3	3.726	312.3	877.2	1189.6	120
125	344.4	3.583	315.5	874.7	1190.3	125
130	347.4	3.452	318.6	872.3	1191.0	130
135	350.3	3.331	321.7	869.9	1191.6	135
140	353.1	3.219	324.6	867.6	1192.2	140
145	355.8	3.112	327.4	865.4	1192.8	145
150	358.5	3.012	330.2	863.2	1193.4	150
155	361.0	2.920	332.9	861.0	1194.0	155
160	363.6	2.834	335.6	858.8	1194.5	160
165	366.0	2.753	338.2	856.8	1195.0	165
170	368.5	2.675	340.7	854.7	1195.4	170
175	370.8	2.602	343.2	852.7	1195.9	175
180	373.1	2.533	345.6	850.8	1196.4	180
185	375.4	2.468	348.0	848.8	1196.8	185
190	377.6	2.406	350.4	846.9	1197.3	190
195	379.8	2.346	352.7	845.0	1197.7	195
200	381.9	2.290	354.9	843.2	1198.1	200
205	384.0	2.237	357.1	841.4	1198.5	205
210	386.0	2.187	359.2	839.6	1198.8	210
215	388.0	2.138	361.4	837.9	1199.2	215
220	389.9	2.091	363.4	836.2	1199.6	220
225	391.9	2.046	365.5	834.4	1199.9	225
230	393.8	2.004	367.5	832.8	1200.2	230
235	395.6	1.964	369.4	831.1	1200.6	235
240	397.4	1.924	371.4	829.5	1200.9	240
245	399.3	1.887	373.3	827.9	1201.2	245
250	401.1	1.850	375.2	826.3	1201.5	250

Table 28-22. Indicated Horsepower of an Engine

A = area of the piston in square inches. P = mean effective pressure of the steam on the piston, lb. per sq. in. L = length of stroke in ft. N = number of working strokes per min. = 2 × r. p. m. for double-acting cylinder.

$$\text{Then i.hp.} = \frac{\text{PLAN}}{33{,}000}$$

The mean pressure in the cylinder of a non-condensing engine when cutting off at

¼ stroke = boiler pressure multiplied by .597	⅝ stroke = boiler pressure multiplied by .919
⅓ " = " " " " .670	⅔ " = " " " " .937
⅜ " = " " " " .743	¾ " = " " " " .966
½ " = " " " " .817	⅞ " = " " " " .992

Table 28-23. Dimensions of Horizontal Return Tubular Boilers* Corresponding to Am. Soc. M. E. Standards

Horse power †	Heating surface Sq. ft.	Shell Dia. In.	Shell Lgth Feet	Tubes No. ‡	Tubes Dia. In.	Tubes Lgth Feet	125-Lb. working pressure Shell In.	125-Lb. Heads In.	125-Lb. Long joint	150-Lb. working pressure Shell In.	150-Lb. Hds In.	150-Lb. Long joint	Diameter of nozzle, in.	Diameter of feed pipe, in.	Diameter of blow-off, in.
34	370	42	12	34	3	12	5/16	3/8	Double Butt.	11/32	7/16	Triple Butt.	4	1	2
36	430	42	14	34	3	14	5/16	3/8	Double Butt.	11/32	7/16	Triple Butt.	4	1	2
39	470	48	12	44	3	12	11/32	½	Double Butt.	3/8	½	Triple Butt.	4	1	2
36	430	48	12	34	3½	12	11/32	½	Double Butt.	3/8	½	Triple Butt.	4	1	2
30	360	48	12	24	4	12	11/32	½	Double Butt.	3/8	½	Triple Butt.	4	1	2
45	540	48	14	44	3	14	11/32	½	Double Butt.	3/8	½	Triple Butt.	4	1	2
42	500	48	14	34	3½	14	11/32	½	Double Butt.	3/8	½	Triple Butt.	4	1	2
35	420	48	14	24	4	14	11/32	½	Double Butt.	3/8	½	Triple Butt.	4	1	2
52	620	48	16	44	3	16	11/32	½	Double Butt.	3/8	½	Triple Butt.	4	1	2
48	570	48	16	34	3½	16	11/32	½	Double Butt.	3/8	½	Triple Butt.	4	1	2
40	480	48	16	24	4	16	11/32	½	Double Butt.	3/8	½	Triple Butt.	4	1	2
47	560	54	12	54	3	12	3/8	½	Triple Butt.	7/16	9/16	Triple Butt.	4	1¼	2
45	540	54	12	44	3½	12	3/8	½	Triple Butt.	7/16	9/16	Triple Butt.	4	1¼	2
43	510	54	12	36	4	12	3/8	½	Triple Butt.	7/16	9/16	Triple Butt.	4	1¼	2
55	660	54	14	54	3	14	3/8	½	Triple Butt.	7/16	9/16	Triple Butt.	4	1¼	2
53	630	54	14	44	3½	14	3/8	½	Triple Butt.	7/16	9/16	Triple Butt.	4	1¼	2
50	600	54	14	36	4	14	3/8	½	Triple Butt.	7/16	9/16	Triple Butt.	4	1¼	2
63	750	54	16	54	3	16	3/8	½	Triple Butt.	7/16	9/16	Triple Butt.	4	1¼	2
60	720	54	16	44	3½	16	3/8	½	Triple Butt.	7/16	9/16	Triple Butt.	4	1¼	2
58	700	54	16	36	4	16	3/8	½	Triple Butt.	7/16	9/16	Triple Butt.	4	1¼	2
85	1021	60	16	76	3	16	3/8	9/16	Quad Butt.	29/64	9/16	Quad Butt.	5	1½	2½
73	872	60	16	54	3½	16	3/8	9/16	Quad Butt.	29/64	9/16	Quad Butt.	5	1½	2½
68	822	60	16	44	4	16	3/8	9/16	Quad Butt.	29/64	9/16	Quad Butt.	5	1½	2½
96	1147	60	18	76	3	18	3/8	9/16	Quad Butt.	29/64	9/16	Quad Butt.	5	1½	2½
82	980	60	18	54	3½	18	3/8	9/16	Quad Butt.	29/64	9/16	Quad Butt.	5	1½	2½
77	924	60	18	44	4	18	3/8	9/16	Quad Butt.	29/64	9/16	Quad Butt.	5	1½	2½
111	1338	66	16	102	3	16	7/16	5/8	Quad Butt.	½	5/8	Quad Butt.	6	2	2½
95	1132	66	16	72	3½	16	7/16	5/8	Quad Butt.	½	5/8	Quad Butt.	6	2	2½
83	993	66	16	54	4	16	7/16	5/8	Quad Butt.	½	5/8	Quad Butt.	6	2	2½
125	1504	66	18	102	3	18	7/16	5/8	Quad Butt.	½	5/8	Quad Butt.	6	2	2½
106	1272	66	18	72	3½	18	7/16	5/8	Quad Butt.	½	5/8	Quad Butt.	6	2	2½
93	1116	66	18	54	4	18	7/16	5/8	Quad Butt.	½	5/8	Quad Butt.	6	2	2½
136	1632	72	16	126	3	16	29/64	5/8	Quad Butt.	17/32	5/8	Quad Butt.	6	2	2½
123	1474	72	16	96	3½	16	29/64	5/8	Quad Butt.	17/32	5/8	Quad Butt.	6	2	2½
107	1289	72	16	72	4	16	29/64	5/8	Quad Butt.	17/32	5/8	Quad Butt.	6	2	2½
153	1834	72	18	126	3	18	29/64	5/8	Quad Butt.	17/32	5/8	Quad Butt.	6	2	2½
138	1657	72	18	96	3½	18	29/64	5/8	Quad Butt.	17/32	5/8	Quad Butt.	6	2	2½
120	1448	72	18	72	4	18	29/64	5/8	Quad Butt.	17/32	5/8	Quad Butt.	6	2	2½
169	2037	72	20	126	3	20	29/64	5/8	Quad Butt.	17/32	5/8	Quad Butt.	6	2	2½
153	1839	72	20	96	3½	20	29/64	5/8	Quad Butt.	17/32	5/8	Quad Butt.	6	2	2½
134	1608	72	20	72	4	20	29/64	5/8	Quad Butt.	17/32	5/8	Quad Butt.	6	2	2½
178	2139	78	18	148	3	18	½	5/8	Quad Butt.	19/32	5/8	Quad Butt.	7	2	2½
167	2001	78	18	118	3½	18	½	5/8	Quad Butt.	19/32	5/8	Quad Butt.	7	2	2½
145	1745	78	18	88	4	18	½	5/8	Quad Butt.	19/32	5/8	Quad Butt.	7	2	2½
197	2375	78	20	148	3	20	½	5/8	Quad Butt.	19/32	5/8	Quad Butt.	7	2	2½
186	2232	78	20	118	3½	20	½	5/8	Quad Butt.	19/32	5/8	Quad Butt.	7	2	2½
161	1938	78	20	88	4	20	½	5/8	Quad Butt.	19/32	5/8	Quad Butt.	7	2	2½

*Coatesville Boiler Works, Philadelphia, Pa.

†For heating boilers, a boiler horsepower is assumed in this table to be equivalent to 12 sq. ft. of heating surface

‡A boiler of 48-in. diameter and larger has a manhole in the front head below the tubes in addition to the regular manhole in the upper part of the shell or front head

Table 28-24. Properties of Air

Temperature, deg. fahr.	Vol. of dry air with unity at 32 deg. fahr.	Cubic feet per lb. of air	Weight per cu. ft. of dry air in lb.	Elastic force of vapor in in. of mercury	Cubic feet of vapor from 1 lb. of water	B.t.u. absorbed per cu. ft. of air per deg. fahr.		Cu. ft. of air raised 1 deg. fahr. by 1 b.t.u.	
						Dry air	Sat. air	Dry air	Sat. air
Zero	0.935	11.58	0.0864	0.044	......	0.02056	0.02054	48.5	48.7
12	0.960	11.87	0.0842	0.074	......	0.02004	0.02006	50.1	50.0
22	0.980	12.14	0.0824	0.118	......	0.01961	0.01963	51.1	51.0
32	1.000	12.40	0.0807	0.181	3289	0.01921	0.01924	52.0	51.8
42	1.020	12.64	0.0791	0.267	2252	0.01882	0.01881	53.2	52.8
52	1.041	12.88	0.0776	0.388	1595	0.01847	0.01848	54.0	53.8
60	1.057	12.39	0.0764	0.522	1227	0.01818	0.01822	55.0	54.6
62	1.061	13.13	0.0761	0.556	1135	0.01811	0.01812	55.2	54.7
70	1.078	13.34	0.0750	0.754	882	0.01777	0.01794	56.3	55.5
72	1.082	13.39	0.0747	0.785	819	0.01767	0.01790	56.5	55.8
82	1.102	13.64	0.0733	1.092	600	0.01744	0.01770	57.2	56.5
92	1.122	13.90	0.0720	1.501	444	0.01710	0.01751	58.5	57.1
100	1.139	13.95	0.0710	1.929	356	0.01690	0.01735	59.1	57.8
102	1.143	14.14	0.0707	2.036	334	0.01682	0.01731	59.5	57.8
112	1.163	14.40	0.0694	2.731	253	0.01651	0.01711	60.6	58.5
122	1.184	14.65	0.0682	3.621	194	0.01623	0.01691	61.7	59.1
132	1.204	14.90	0.0671	4.752	151	0.01596	0.01670	62.5	59.9
142	1.224	15.15	0.0660	6.165	118	0.01571	0.01652	63.7	60.6
152	1.245	15.40	0.0649	7.930	93.3	0.01544	0.01634	65.0	61.5
162	1.265	15.65	0.0638	10.099	74.5	0.01518	0.01616	66.2	62.4
172	1.285	15.90	0.0628	12.758	59.2	0.01494	0.01598	67.1	63.3
182	1.306	16.17	0.0618	15.960	48.6	0.01471	0.01580	68.0	64.2
192	1.326	16.42	0.0609	19.828	39.8	0.01449	....	68.9	....
202	1.347	16.67	0.0600	24.450	32.7	0.01426	....	69.5	....
212	1.367	16.92	0.0591	29.921	27.1	0.01406	....	71.4	....

Table 28-25. Volume and Weight of Air at Atmospheric Pressure at Temperatures Between 212 and 850 Deg. Fahr.

Temperature, degrees fahrenheit	Volume of one pound in cubic feet	Weight one cubic foot in pounds	Temperature, degrees fahrenheit	Volume of one pound in cubic feet	Weight one cubic foot in pounds	Temperature, degrees fahrenheit	Volume of one pound in cubic feet	Weight one cubic foot in pounds
212	16.925	.059084	320	19.647	.050898	550	25.444	.039302
220	17.127	.058388	340	20.151	.049625	575	26.074	.038352
230	17.379	.057541	360	20.655	.048414	600	26.704	.037448
240	17.631	.056718	380	21.159	.047261	650	27.964	.035760
250	17.883	.055919	400	21.663	.046162	700	29.224	.034219
260	18.135	.055142	425	22.293	.044857	750	30.484	.032804
270	18.387	.054386	450	22.923	.043624	800	31.744	.031502
280	18.639	.053651	475	23.554	.042456	850	33.004	.030299
290	18.891	.052935	500	24.184	.041350			
300	19.143	.052238	525	24.814	.040300			

Table 28-26. Weight of Water at Temperatures Used in Physical Calculations

Temperature, Degrees Fahrenheit	Weight per cubic foot, pounds	Weight per cubic inch, pounds
At 32 degrees or freezing point at sea level	62.418	0.03612
At 39.2 degrees or point of maximum density	62.427	0.03613
At 62 degrees or standard temperature	62.355	0.03608
At 212 degrees or boiling point at sea level	59.846	0.03469

Table 28-27. Volume and Weight of Distilled Water at Various Temperatures*

Temperature, deg. fahr.	Relative volume water at 39.2 deg.=1	Weight in lb. per cubic foot	Temperature, deg. fahr.	Relative volume, water at 39.2 deg.=1	Weight in lb. per cubic foot	Temperature, deg. fahr.	Relative volume, water at 39.2 deg.=1	Weight in lb. per cubic foot	Temperature, deg. fahr.	Relative volume, water at 39.2 deg. =1	Weight in lb. per cubic foot
32	1.000176	62.42	160	1.02337	61.00	290	1.0830	57.65	430	1.197	52.2
39.2	1.000000	62.43	170	1.02682	60.80	300	1.0890	57.33	440	1.208	51.7
40	1.000004	62.43	180	1.03017	60.58	310	1.0953	57.00	450	1.220	51.2
50	1.00027	62.42	190	1.03431	60.36	320	1.1019	56.66	460	1.232	50.7
60	1.00096	62.37	200	1.03835	60.12	330	1.1088	56.30	470	1.244	50.2
70	1.00201	62.30	210	1.04256	59.88	340	1.1160	55.94	480	1.256	49.7
80	1.00338	62.22	212	1.04343	59.83	350	1.1235	55.57	490	1.269	49.2
90	1.00504	62.11	220	1.0469	59.63	360	1.1313	55.18	500	1.283	48.7
100	1.00698	62.00	230	1.0515	59.37	370	1.1396	54.78	510	1.297	48.1
110	1.00915	61.86	240	1.0562	59.11	380	1.1483	54.36	520	1.312	47.6
120	1.01157	61.71	250	1.0611	58.83	390	1.1573	53.94	530	1.329	47.0
130	1.01420	61.55	260	1.0662	58.55	400	1.167	53.5	540	1.35	46.3
140	1.01705	61.38	270	1.0715	58.26	410	1.177	53.0	550	1.37	45.6
150	1.02011	61.20	280	1.0771	57.96	420	1.187	52.6	560	1.39	44.9

* Marks and Davis.

Table 28-28. Boiling Point of Water at Various Altitudes

Boiling point, degrees fahrenheit	Altitude above sea level, feet	Atmospheric pressure, pounds per square inch	Barometer reduced to 32 degrees, inches	Boiling point, degrees fahrenheit	Altitude above sea level, feet	Atmospheric pressure, pounds per square inch	Barometer reduced to 32 degrees, inches
184	15221	8.20	16.70	199	6843	11.29	22.99
185	14649	8.38	17.06	200	6304	11.52	23.47
186	14075	8.57	17.45	201	5764	11.76	23.95
187	13498	8.76	17.83	202	5225	12.01	24.45
188	12934	8.95	18.22	203	4697	12.26	24.96
189	12367	9.14	18.61	204	4169	12.51	25.48
190	11799	9.34	19.02	205	3642	12.77	26.00
191	11243	9.54	19.43	206	3115	13.03	26.53
192	10685	9.74	19.85	207	2589	13.30	27.08
193	10127	9.95	20.27	208	2063	13.57	27.63
194	9579	10.17	20.71	209	1539	13.85	28.19
195	9031	10.39	21.15	210	1025	14.13	28.76
196	8481	10.61	21.60	211	512	14.41	29.33
197	7932	10.83	22.05	212	Sea Level	14.70	29.92
198	7381	11.06	22.52				

Table 28-29. Pressures Corresponding to Given Heads of Water in Feet

Water at maximum density. Temperature, 39.2 deg. fahr.
h = head in feet. P = pressure in lb. per sq. inch = .433 h

h	P	h	P	h	P	h	P	h	P	h	P	h	P
1	.433	16	6.928	31	13.42	46	19.92	61	26.41	76	32.91	91	39.40
2	.866	17	7.361	32	13.86	47	20.35	62	26.85	77	33.34	92	39.84
3	1.299	18	7.794	33	14.29	48	20.78	63	27.28	78	33.77	93	40.27
4	1.732	19	8.227	34	14.72	49	21.22	64	27.71	79	34.21	94	40.70
5	2.165	20	8.660	35	15.15	50	21.65	65	28.14	80	34.64	95	41.13
6	2.598	21	9.09	36	15.59	51	22.08	66	28.58	81	35.07	96	41.57
7	3.031	22	9.53	37	16.02	52	22.52	67	29.01	82	35.51	97	42.00
8	3.464	23	9.96	38	16.45	53	22.95	68	29.44	83	35.94	98	42.43
9	3.897	24	10.39	39	16.89	54	23.38	69	29.88	84	36.37	99	42.87
10	4.330	25	10.82	40	17.32	55	23.81	70	30.31	85	36.80	100	43.30
11	4.763	26	11.26	41	17.75	56	24.25	71	30.74	86	37.24		
12	5.196	27	11.69	42	18.19	57	24.68	72	31.18	87	37.67		
13	5.629	28	12.12	43	18.62	58	25.11	73	31.61	88	38.10		
14	6.062	29	12.56	44	19.05	59	25.55	74	32.04	89	38.54		
15	6.495	30	12.99	45	19.48	60	25.98	75	32.47	90	38.97		

Table 28-30. Pressure, in Ounces Per Square Inch Corresponding to Various Heads of Water, in Inches*

Head in inches	Decimal parts of an inch .0	.1	.2	.3	.4	.5	.6	.7	.8	.9
0	....	.06	.12	.17	.23	.29	.35	.40	.46	.52
1	.58	.63	.69	.75	.81	.87	.93	.98	1.04	1.09
2	1.16	1.21	1.27	1.33	1.39	1.44	1.50	1.56	1.62	1.67
3	1.73	1.79	1.85	1.91	1.96	2.02	2.08	2.14	2.19	2.25
4	2.31	2.37	2.42	2.48	2.54	2.60	2.66	2.72	2.77	2.83
5	2.89	2.94	3.00	3.06	3.12	3.18	3.24	3.29	3.35	3.41
6	3.47	3.52	3.58	3.64	3.70	3.75	3.81	3.87	3.92	3.98
7	4.04	4.10	4.16	4.22	4.28	4.33	4.39	4.45	4.50	4.56
8	4.62	4.67	4.73	4.79	4.85	4.91	4.97	5.03	5.08	5.14
9	5.20	5.26	5.31	5.37	5.42	5.48	5.54	5.60	5.66	5.72

*Suplee's *Mechanical Engineers' Reference Book*, published by J. B. Lippincott Co.

Table 28-31. Comparison of Measures of Pressure and Weight†

1 lb. per sq. in. =	144 lb. per sq. ft. 2.0416 in. mercury at 62 deg. fahr. 2.309 ft. water at 62 deg. fahr. 27.71 in. water at 62 deg. fahr.
1 oz. per sq. in. =	0.1276 in. mercury at 62 deg. fahr. 1.732 in. water at 62 deg. fahr.
1 atmosphere (14.7 lb. per sq. in.) =	2116.3 lb. per sq. ft. 33.947 ft. water at 62 deg. fahr. 30 in. mercury at 62 deg. fahr. 29.922 in. mercury at 32 deg. fahr.
1 in. water at 62 deg. fahr. =	0.03609 lb. or .5774 oz. per sq. in. 5.196 lb. per sq. ft.
1 ft. water at 62 deg. fahr. =	0.433 lb. per sq. in. 62.355 lb. per sq. ft.
1 in. mercury at 62 deg. fahr. =	0.491 lb. or 7.86 oz. per sq. in. 1.132 ft. water at 62 deg. fahr. 13.58 in. water at 62 deg. fahr.

†Kent's *Mechanical Engineers' Pocket Book*

Table 28-32. Conversion of Mercury and Vapor Pressures

Inches of mercury to pounds per square inch

Tenths	0	1	2	3	4	5	6	7	8	9
Inches	Lb. Sq. in.	Lb. Sq. in.	Lb. Sq. in.	Lb. Sq. in.	Lb. Sq. in.	Lb. Sq. in.	Lb. Sq. in.	Lb. Sq. in.	Lb. Sq. in.	Lb. Sq. in.
0	0.	0.49	0.98	1.47	1.96	2.46	2.95	3.44	3.93	4.42
10	4.91	5.40	5.89	6.39	6.88	7.37	7.86	8.35	8.84	9.33
20	9.82	10.32	10.81	11.30	11.79	12.28	12.77	13.26	13.75	14.24
30	14.74	15.2	15.7	16.2	16.7	17.2	17.7	18.2	18.7	19.1
40	19.6	20.1	20.6	21.1	21.6	22.1	22.6	23.1	23.6	24.1
50	24.6	25.1	25.5	26.0	26.5	27.0	27.5	28.0	28.5	29.0
60	29.5	30.0	30.5	30.9	31.4	31.9	32.4	32.9	33.4	33.9
70	34.4	34.9	35.4	35.9	36.3	36.8	37.3	37.8	38.3	38.8
80	39.3	39.8	40.3	40.8	41.3	41.8	42.2	42.7	43.2	43.7
90	44.2	44.7	45.2	45.7	46.2	46.7	47.2	47.6	48.1	48.6
100	49.1	49.6	50.1	50.6	51.1	51.6	52.1	52.6	53.0	53.5

Pounds per square inch to inches of mercury

Tenths	0	1	2	3	4	5	6	7	8	9
Pounds	In. Hg.	In. Hg.	In. Hg.	In. Hg.	In. Hg.	In. Hg.	In. Hg.	In. Hg.	In. Hg.	In. Hg.
0	0.	2.0352	4.0704	6.1056	8.1408	10.1760	12.2112	14.2464	16.2816	18.3168
10	20.352	22.3872	24.4224	26.4576	28.4928	30.528	32.5632	34.5984	36.6336	38.6688
20	40.704	42.7392	44.7744	46.8096	48.8448	50.8809	52.9152	54.9504	56.9856	59.0208
30	61.056	63.0912	65.1264	67.1616	69.1968	71.2320	73.2672	75.3024	77.3376	79.3728
40	81.408	83.4432	85.4784	87.5136	89.5488	91.5810	93.6192	95.6544	97.6896	99.7148
50	101.76	103.795	105.830	107.865	109.900	111.936	113.971	116.006	118.041	120.077
60	122.11	124.145	126.180	128.215	130.250	132.286	134.321	136.356	138.391	140.427
70	142.46	144.495	146.530	148.565	150.600	152.636	154.671	156.706	158.741	160.777
80	162.81	164.945	166.880	168.915	170.950	172.986	175.021	177.056	179.091	181.127
90	183.16	185.195	187.230	189.265	191.300	193.336	195.371	197.406	199.441	201.476
100	203.53	205.565	207.600	209.635	211.670	213.706	215.741	217.776	219.811	221.846

Table 28-33. Comparison of Measures of Pressure

Name of units	Atmospheres	On square inch	Inches mercury at 32 deg. fahr.	Feet of water at 60 deg. fahr.	Millimeters of mercury at 32° fahr.	Pounds per square foot	Kilograms per square meter
Atmosphere	1.	14.7	29.922	33.94	760.	2,116.	10,333
Pounds per square inch	.068,03	1.	2.036	2.309	51.7	143.946	702.925
In. mercury at 32° fahr.	.033,42	.491,3	1.	1.134	25.398	70.7	345.331
Feet of water at 60° fahr.	.029,47	.433,2	.881,8	1.	22.399	62.35	304.565
Millimeters of mercury at 32° fahr.	.001,316	.019,34	.039,37	.044,64	1.	2.784	13.596
Pounds per square foot	.000,472,6	.006,947	.014,13	.016,03	.359,2	1.	4.883
Kilograms per sq. meter	.000,096,77	.001,423	.002,895	.003,283	.073,55	.204,8	1.

Table 28-34. Reasonable Economic Performance of Stationary Steam Plants*

Type of plant	Central station		Mfg. power plants			Heating plants		
	Large 10,000 kw. and up	Small 2000-10,000 kw.	Small up to 100 hp.	Medium 100-500 hp.	Large 500-2000 hp.	Central 1000 hp. and up	Office and public bldgs.	Residence
Efficiency of boiler and Furnace in per cent	70-76	68-74	60-70	68-72	68-74	68-74	50-70	50-65
Coal per hour in lb.	Per kw-hr.		Per 1 hp.			Per boiler hp.		
	2-3	2½-4	5-8	3-5	2½-4	3-4	3-6	

* L. P. Breckenridge. Lecture on Fuel Conservation

Table 28-35. Weight in Pounds of One Gallon of Water at Temperatures from 32 Deg. to 420 Deg. Fahr.

Temp.	Wt.	Temp.	Wt.	Temp.	Wt.	Temp.	Wt.
32	8.344	105	8.279	185	8.084	270	7.788
35	8.345	110	8.270	190	8.069	280	7.748
39.2	8.3454	115	8.260	195	8.053	290	7.707
40	8.345	120	8.250	200	8.037	300	7.664
45	8.345	125	8.239	205	8.021	310	7.620
50	8.343	130	8.229	210	8.005	320	7.575
55	8.311	135	8.218	212	7.998	330	7.527
60	8.337	140	8.206	215	7.988	340	7.486
65	8.333	145	8.193	220	7.971	350	7.429
70	8.329	150	8.181	225	7.954	360	7.376
75	8.323	155	8.168	230	7.937	370	7.323
80	8.317	160	8.155	235	7.929	380	7.267
85	8.311	165	8.141	240	7.920	390	7.211
90	8.304	170	8.127	245	7.893	400	7.152
95	8.296	175	8.113	250	7.865	410	7.085
100	8.288	180	8.099	260	7.828	420	7.032

Table 28-36. Contents of Round Tanks in U. S. Gallons, for Each Foot in Depth

To find capacity of a tank of any size: Given dimensions of a cylinder in inches, to find its capacity in U. S. gallons: Square the diameter, multiply by the length and by .0034

Diameter Ft.	Diameter In.	Gallons, 1 foot in depth	Diameter Ft.	Diameter In.	Gallons, 1 foot in depth	Diameter Ft.	Diameter In.	Gallons, 1 foot in depth	Diameter Ft.	Diameter In.	Gallons, 1 foot in depth
1	0	5.8735	7	0	287.8032	15	0	1321.5454	23	0	3107.1001
1	3	9.1766	7	3	308.7270	15	3	1365.9634	23	3	3175.0122
1	6	13.2150	7	6	330.3859	15	6	1407.5165	23	6	3243.6595
1	9	17.9870	7	9	352.7665	15	9	1457.0032	23	9	3313.0103
2	0	23.4940	8	0	375.9062	16	0	1503.6250	24	0	3383.1563
2	3	29.7340	8	3	399.7666	16	3	1550.9797	24	3	3454.0051
2	6	36.7092	8	6	424.3625	16	6	1599.0696	24	6	3525.5929
2	9	44.4179	8	9	449.2118	16	9	1647.8930	24	9	3597.9068
3	0	52.8618	11	0	710.6977	17	0	1697.4516	25	0	3670.9596
3	3	62.0386	11	3	743.3686	17	3	1747.7431	25	3	3744.7452
3	6	73.1504	11	6	776.7746	17	6	1798.7698	25	6	3819.2657
3	9	82.5959	11	9	810.9143	17	9	1850.5301	25	9	3894.5203
4	0	93.9754	12	0	848.1890	18	0	1903.0254	26	0	3970.5098
4	3	106.1200	12	3	881.3966	18	3	1956.2537	26	3	4047.2322
4	6	118.9386	12	6	917.7395	18	6	2010.2171	26	6	4124.6898
4	9	132.5209	12	9	954.8159	18	9	2064.9140	26	9	4202.9610
5	0	146.8384	13	0	992.6274	21	0	2590.2290	27	0	4281.8072
5	3	161.8886	13	3	1031.1719	21	3	2652.2532	27	3	4361.4664
5	6	177.6740	13	6	1070.4514	21	6	2715.0413	27	6	4441.8607
5	9	194.1913	13	9	1108.0645	21	9	2778.5486	27	9	4522.9886
6	0	211.4472	14	0	1151.2129	22	0	2842.7910	28	0	4604.8517
6	3	229.4342	14	3	1192.6940	22	3	2907.7664	28	3	4686.4876
6	6	248.1564	14	6	1234.9104	22	6	2973.4889	28	6	4770.7787
6	9	267.6122	14	9	1277.8615	22	9	3039.9209	28	9	4854.8434

Table 28-37. Friction of Water in Elbows *

Loss of head in feet, due to friction in various sizes of smooth 90-deg. elbows when discharging the given quantities of water

If pipe is slightly rough add 15 per cent If very rough add 30 per cent
Vel.—Velocity in feet per second Fric.—Friction head in feet

Table shows loss for one elbow, and is based on Weisbach's formula for short radius bends
*Yeoman Brothers Company

Gal'ns per minute	1 Inch		1¼ Inches		1½ Inches		2 Inches		2½ Inches		3 Inches		4 Inches		5 Inches		6 Inches		8 Inches		10 Inches		12 Inches	
	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.
5	2.04	0.06	1.30	0.02	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
10	4.08	0.22	2.60	0.08	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
15	6.12	0.49	3.90	0.19	2.73	0.09	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
20	8.16	0.87	5.20	0.33	3.64	0.16	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
25	10.20	1.35	6.50	0.52	4.55	0.25	2.60	0.09	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
30	12.24	1.95	7.80	0.76	5.46	0.36	3.06	0.13	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
35	14.28	2.65	9.10	1.01	6.37	0.50	3.57	0.18	2.29	0.09	....	....	....	....	....	....	....	....	....	....	....	....	....	....
40	16.32	3.46	10.40	1.36	7.28	0.64	4.05	0.23	2.62	0.11	....	....	....	....	....	....	....	....	....	....	....	....	....	....
45	....	....	11.70	1.71	8.19	0.81	4.60	0.29	2.95	0.14	2.02	0.06	....	....	....	....	....	....	....	....	....	....	....	....
50	....	....	....	....	9.10	0.99	5.11	0.35	3.30	0.18	2.27	0.08	....	....	....	....	....	....	....	....	....	....	....	....
70	....	....	....	....	12.74	1.98	7.15	0.70	4.60	0.34	3.18	0.19	1.79	0.05	....	....	....	....	....	....	....	....	....	....
100	....	....	....	....	....	....	10.20	1.41	6.51	0.74	4.54	0.29	2.55	0.10	....	....	....	....	....	....	....	....	....	....
120	....	....	....	....	....	....	12.25	2.24	7.81	1.17	5.45	0.50	3.06	0.15	1.96	0.06	....	....	....	....	....	....	....	....
150	....	....	....	....	....	....	15.30	3.20	9.80	1.58	6.80	0.66	3.84	0.22	2.45	0.09	....	....	....	....	....	....	....	....
175	....	....	....	....	....	....	....	....	11.43	2.16	7.92	0.90	4.45	0.30	2.86	0.12	2.00	0.06	....	....	....	....	....	....
200	....	....	....	....	....	....	....	....	13.07	2.96	9.08	1.18	5.11	0.40	3.27	0.16	2.28	0.07	....	....	....	....	....	....
250	....	....	....	....	....	....	....	....	....	....	11.28	1.84	6.40	0.62	4.08	0.25	2.80	0.12	1.60	0.01	....	....	....	....
270	....	....	....	....	....	....	....	....	....	....	12.45	2.35	6.90	0.70	4.42	0.27	3.03	0.14	1.70	0.01	....	....	....	....
300	....	....	....	....	....	....	....	....	....	....	13.62	2.63	7.66	0.89	4.90	0.36	3.40	0.18	1.90	0.06	....	....	....	....
350	....	....	....	....	....	....	....	....	....	....	....	....	8.90	1.24	5.72	0.50	3.98	0.24	2.20	0.09	....	....	....	....
400	....	....	....	....	....	....	....	....	....	....	....	....	10.20	1.59	6.54	0.63	4.54	0.29	2.60	0.10	....	....	....	....
450	....	....	....	....	....	....	....	....	....	....	....	....	11.50	2.01	7.35	0.81	5.12	0.39	2.92	0.13	1.80	0.05	....	....
470	....	....	....	....	....	....	....	....	....	....	....	....	12.16	2.26	7.78	0.90	5.49	0.46	3.07	0.14	1.92	0.06	....	....
500	....	....	....	....	....	....	....	....	....	....	....	....	12.77	2.47	8.17	1.01	5.60	0.48	3.20	0.16	2.00	0.06	1.40	0.04
750	....	....	....	....	....	....	....	....	....	....	....	....	....	....	12.26	2.24	8.40	1.09	4.80	0.36	3.00	0.15	2.10	0.07
1050	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	12.57	2.41	7.04	0.76	4.40	0.29	3.08	0.14
1250	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	14.10	3.02	8.00	1.00	5.00	0.40	3.50	0.20
1500	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	9.60	1.44	6.10	0.58	4.20	0.29
2000	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	12.70	2.44	8.10	1.01	5.60	0.47
2500	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	15.90	3.68	10.10	1.57	7.00	0.75
3000	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	12.10	2.25	8.40	1.08
3500	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	14.10	3.05	9.80	1.47
4000	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	11.35	1.92
4200	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	11.93	2.18
4500	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	12.78	2.43
5000	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	14.20	3.01

Table 28-38. Friction of Water in Pipes

Giving velocity in feet per second, friction head in feet and friction loss in pounds per square inch for each 100 ft. of pipe discharging a given quantity of water in gallons per minute (Weisbach Formula)

Gallons per minute	Velocity in ft. per second	Friction head in feet	Friction loss lb. per sq. in.	Velocity in ft. per second	Friction head in feet	Friction loss lb. per sq. in.	Velocity in ft. per second	Friction head in feet	Friction loss lb. per sq. in.	Velocity in ft. per second	Friction head in feet	Friction loss lb. per sq. in.	Velocity in ft. per second	Friction head in feet	Friction loss lb. per sq. in.	Velocity in ft. per second	Friction head in feet	Friction loss lb. per. sq. in.
	¾″ Pipe			1″ Pipe			1¼″ Pipe			1½″ Pipe			2″ Pipe			2½″ Pipe		
5	3.64	7.59	3.3	2.04	1.93	0.84	1.30	0.71	0.31	0.91	0.27	0.12	0.49	0.092	0.04	0.244	0.046	0.02
10	7.28	29.90	13.0	4.08	10.26	3.16	2.60	2.41	1.05	1.82	1.08	0.47	0.98	0.277	0.12	0.656	0.092	0.04
15	10.92	66.01	28.7	6.12	16.05	6.98	3.90	5.47	2.38	2.73	2.23	0.97	1.47	0.577	0.25	0.985	0.185	0.08
20	14.56	115.92	50.4	8.16	28.29	12.30	5.20	9.36	4.07	3.64	3.81	1.66	2.04	0.97	0.42	1.315	0.323	0.14
25	18.20	180.00	78.00	10.20	43.70	19.00	6.50	14.72	6.4	4.55	5.02	2.62	2.60	1.43	0.62	1.645	0.485	0.21
30				12.24	63.25	27.50	7.80	21.04	9.15	5.46	8.62	3.75	3.03	2.09	0.91	1.97	0.693	0.30
35				14.28	85.10	37.00	9.10	28.52	12.4	6.37	11.61	5.05	3.54	2.76	1.22	2.29	0.92	0.40
40				16.32	110.40	48.00	10.40	37.03	16.10	7.28	14.99	6.52	4.05	3.68	1.60	2.62	1.19	0.53
45							11.70	46.46	20.2	8.19	18.74	8.15	4.56	4.60	1.99	2.95	1.49	0.66
50							13.00	57.27	24.9	9.10	23.00	10.00	5.10	5.61	2.44	3.30	1.86	0.81
60							15.6	85.50	37.0	10.92	32.95	14.25	6.12	8.88	3.50	3.95	2.70	1.17
70							18.2	114.0	49.3	12.74	44.60	19.30	7.14	11.09	4.80	4.60	3.46	1.50
75							19.5	129.0	56.1	13.65	51.52	22.4	7.70	12.23	5.32	4.93	4.14	1.80
80										14.56	58.45	25.3	8.16	14.55	6.30	5.26	4.62	2.00
90										16.38	81.50	35.25	9.18	18.02	7.80	5.91	5.96	2.58
100										18.20	89.70	39.0	10.2	21.75	9.46	6.50	7.36	3.20
125													12.80	34.27	14.9	8.13	11.24	4.89
150													15.3	48.76	21.2	9.80	16.10	7.00
175																11.43	21.75	9.46
185																12.08	24.50	10.61
200																13.06	28.68	12.47
	3″ Pipe			3½″ Pipe			4″ Pipe			5″ Pipe			6″ Pipe			7″ Pipe		
10	0.448	0.046	0.02															
15	0.672	0.092	0.04	0.498	0.046	0.02												
20	0.896	0.138	0.06	0.664	0.069	0.03												
25	1.12	0.231	0.10	0.83	0.092	0.04												
30	1.345	0.30	0.13	0.996	0.138	0.06												
35	1.569	0.393	0.17	1.163	0.208	0.09												
40	1.790	0.53	0.23	1.329	0.254	0.11	1.04	0.138	0.06									
45	2.016	0.647	0.28	1.494	0.323	0.14	1.17	0.1615	0.07									
50	2.24	0.80	0.35	1.66	0.393	0.17	1.30	0.208	0.09									
60	2.688	1.155	0.50	1.992	0.555	0.24	1.56	0.30	0.13	0.88	0.1156	0.05						
70	3.136	1.385	0.60	2.324	0.879	0.38	1.82	0.439	0.19	1.04	0.162	0.07						
75	3.360	1.70	0.75	2.490	0.913	0.395	1.95	0.485	0.21	1.20	0.174	0.075						
80	3.584	2.08	0.90	2.656	0.948	0.41	2.08	0.580	0.23	1.28	0.185	0.08						
90	4.032	2.54	1.10	2.988	1.247	0.56	2.34	0.60	0.26	1.44	0.208	0.09						
100	4.480	3.01	1.31	3.320	1.478	0.64	2.60	0.763	0.33	1.60	0.277	0.12	1.14	0.115	0.05			
125	5.60	4.57	1.99	4.15	2.219	0.96	3.25	1.13	0.49	2.00	0.393	0.17	1.42	0.161	0.07			
150	5.80	6.55	2.85	4.98	3.12	1.35	3.80	1.59	0.69	2.40	0.578	0.25	1.71	0.231	0.10	1.20	0.093	0.04
175	7.92	8.85	3.85	5.81	4.208	1.82	4.45	2.146	0.93	2.80	0.785	0.34	2.00	0.302	0.13	1.38	0.115	0.05
185	8.34	9.94	4.30	6.14	4.62	2.00	4.70	2 484	1.075	2.96	0.84	0.36	2.11	0.36	0.156	1.55	0.13	0.056
200	9.04	11.54	5.02	6.64	5.50	2.38	5.1	2 82	1.22	3.20	0.972	0.42	2.28	0.39	0.17	1.70	0.162	0.07
250	11.28	17.84	7.76	8.30	8.55	3.70	6.4	4.37	1.89	4.00	1.50	0.65	2.80	0.60	0.26	2.10	0.277	0.12
265	12.40	20.09	8.72	8.80	9.60	4.15	6.79	6.45	2.09	4.24	1.69	0.73	3.03	0.70	0.303	2.23	0.31	0.134
300	13.52	25.76	11.20	9.96	11.63	5.04	7.60	6.15	2.66	4.80	2.15	0.93	3.40	0.85	0.37	2.40	0.393	0.17

Hot water averages 8 lb. per gallon

Horsepower required to raise water: — horsepower = quantity in cu. ft. per min. × height of lift in feet ÷ 529.2 = quantity in gal. per min. × height of lift in feet ÷ 3958.7

When the temperature of water increases, the pressure of the water vapor decreases the theoretical lift, which at 150 deg. fahr. = 25.7 ft.; at 175 deg. = 18.5 ft., and at 200 deg. = 7.2 ft.

Table 28-39. Cost of Water at Stated Rates per 1000 Gallons

Number of cubic feet	Cost per 1000 Gallons							
	5 Cents	6 Cents	8 Cents	10 Cents	15 Cents	20 Cents	25 Cents	30 Cents
20	$0.007	$0.009	$0.012	$0.015	$0.021	$0.030	$0.037	$0.045
40	0.015	0.018	0.024	0.030	0.045	0.060	0.075	0.090
60	0.022	0.027	0.036	0.045	0.066	0.090	0.112	0.135
80	0.030	0.036	0.048	0.060	0.090	0.120	0.150	0.180
100	0.037	0.019	0.060	0.075	0.111	0.150	0.187	0.224
200	0.075	0.090	0.120	0.150	0.225	0.299	0.374	0.449
300	0.112	0.135	0.180	0.224	0.336	0.449	0.561	0.673
400	0.150	0.180	0.239	0.299	0.450	0.598	0.748	0.898
500	0.188	0.224	0.299	0.374	0.564	0.748	0.935	1.122
600	0.224	0.269	0.359	0.449	0.448	0.898	1.122	1.346
700	0.262	0.314	0.419	0.524	0.786	1.047	1.309	1.571
800	0.299	0.350	0.479	0.598	0.897	1.197	1.496	1.795
900	0.337	0.404	0.539	0.673	1.011	1.346	1.683	2.020
1,000	0.374	0.449	0.598	0.748	1.122	1.496	1.870	2.244
2,000	0.748	0.898	1.197	.496	244	2.992	3.740	4.488
3,000	1.122	1.346	1.795	2.244	3.366	4.488	5.610	6.732
4,000	1.496	1.795	2.393	2.992	4.188	5.849	7.480	8.976
5,000	1.870	2.244	2.992	3.740	5.610	7.480	9.350	11.220
6,000	2.241	2.692	3.590	4.488	6.732	8.976	11.220	13.464
7,000	2.618	3.141	4.189	5.236	7.854	10.472	13.090	15.708
8,000	2.992	3.590	4.787	5.984	8.976	11.968	14.961	17.953
9,000	3.366	4.039	5.385	6.732	10.098	13.164	16.831	20.197
10,000	3.74	4.488	5.984	7.480	11.122	14.961	18.701	22.441
20,000	7.48	8.976	11.968	14.961	22.443	29.992	37.402	44.882
30,000	11.22	13.46	17.95	22.44	33.664	44.88	56.10	67.32
40,000	14.96	17.95	23.94	29.92	44.885	59.84	74.81	89.77
50,000	18.70	22.44	29.92	37.40	56.103	74.80	93.50	112.20
60,000	22.44	26.92	35.90	44.88	67.323	89.76	112.20	134.64
70,000	26.18	31.41	41.89	52.36	78.513	104.72	130.90	157.08
80,000	29.92	35.90	47.87	59.84	89.766	119.68	149.61	179.53
90,000	33.66	40.39	53.85	67.32	100.986	134.64	168.31	201.97
100,000	37.40	44.88	59.84	74.80	111.22	149.61	187.01	224.41
200,000	74.81	89.76	119.68	149.61	224.43	299.22	374.02	448.82
300,000	112.20	134.64	179.53	224.41	336.64	448.83	561.03	673.24
400,000	149.61	179.53	239.37	299.22	448.85	598.44	748.05	897.66
500,000	187.01	224.41	299.22	374.02	561.03	748.05	935.06	1122.07
600,000	224.41	269.29	359.06	448.83	673.23	897.66	1122.07	1346.49
700,000	261.81	314.18	418.90	523.63	785.43	1047.27	1309.08	1570.88
800,000	299.22	359.06	478.75	598.44	897.66	1196.88	1496.10	1795.32
900,000	336.62	403.94	538.59	673.24	1009.86	1346.49	1683.11	2019.73
1,000,000	374.02	448.83	598.44	748.05	1122.06	1498.10	1870.12	2244.15

Table 28-40. Water Conversion Factors

U. S. gallons	×	8.33	= pounds.	Cubic feet of water (39.2°)	×	62.427	= pounds.
U. S. gallons	×	0.13368	= cubic ft.	Cubic feet of water (39.2°)	×	7.48	= U.S. gal.
U. S. gallons	×	231.00	= cubic in.	Cubic feet of water (39.2°)	×	0.028	= tons.
U. S. gallons	×	3.78	= liters.	Pounds of water	×	27.72	= cubic in.
Cubic inches of water (39.2°)	×	0.036130	= pounds.	Pounds of water	×	0,01602	= cubic ft.
Cubic inches of water (39.2°)	×	0.001329	= U.S. gal.	Pounds of water	×	0.12	= U.S. gal.
Cubic inches of water (39.2°)	×	0.576384	= ounces.				

Table 28-41. Classification of Coals *

1 cu. ft. of anthracite coal weighs 55 to 66 lb.
1 " " " bituminous " " 50 to 55 lb.
1 " " " semi-bituminous coal weighs 48 to 53 lb.

Name of coal	Percentages of combustible		B.t.u. per pound of combustible
	Fixed carbon	Volatile matter	
Anthracite	97.0 to 92.5	3.0 to 7.5	14,600 to 14,800
Semi-anthracite	92.5 to 87.5	7.5 to 12.5	14,700 to 15,500
Semi-bituminous	87.5 to 75.0	12.5 to 25.0	15,500 to 16,000
Bituminous, East	75.0 to 60.0	25.0 to 40.0	14,800 to 15,300
" West	65.0 to 50.0	35.0 to 50.0	13,500 to 14,800
Lignite	50.0 and under	50.0 and over	11,000 to 13,500

* Harding and Willard, *Mechanical Equipment of Buildings*. Published by John Wiley & Sons

Table 28-42. Names and Sizes of Bituminous or Soft Coal †

For "Domestic" soft coals there are no uniform names and sizes, but they are marketed in the various states under about these classes:

Screenings usually smallest sizes.
Duff goes through 1/8-in. screen.
No. 3 Nut goes through 1¼-in. screen, over ¾-in. screen.
No. 2 Nut goes through 2-in. screen, over 1¼-in. screen.
No. 1 Domestic Nut goes through 3-in. screen, over 1½ or 2-in. screen.
No. 4 Washed goes through ¾-in. screen, over ¼-in. screen.
No. 3 Washed Chestnut goes through 1¼-in. screen, over ¾-in. screen.
No. 2 Washed Stove goes through 2-in. screen, over 1¼-in. screen.
No. 1 Washed Egg goes through 3-in. screen, over 2-in. screen.
No. 3 Roller Screened Nut goes through 1½-in. screen, over 1-in. screen.
No. 2 Roller Screened Nut goes through 2-in. screen, over 1½-in. screen.
No. 1 Roller Screened Nut goes through 3½-in. screen, over 2-in. screen.
Egg goes through 6-in. screen, over 3-in. screen.
Lump or Block goes through 6-in. screen, or over.
Run-of-Mine in fine and large lumps.
Pocahontas Smokeless: generally sized as: Nut, Egg, Lump, and Mine-Run.

† Harding and Willard

Table 28-43. Heat Values of Bituminous Coals ‡

From selected free-burning and caking soft fuels taken from *Bulletin No. 332*, U. S. Geological Survey, and *Bulletin No. 23*, U. S. Bureau of Mines

State	Test No.	Kind of fuel	County	B.t.u. per lb. dry coal
Alabama	375	Soft—caking	Bibb	13,671
Alabama	484	Soft—free burning	Jefferson	14,447
Arkansas	293	Soft—caking	Sebastian	13,705
Arkansas	308	Semi-anthracite—caking	Johnson	14,125
Arkansas	340	Lignite	Quachita	9,549
Georgia	481	Soft—free burning	Chattooga	12,865
Illinois	448	Soft—free burning	Williamson	12,920
Illinois	511	Soft briquettes	St. Clair	13,271
Illinois	509	Soft—caking	Saline	13,621
Indiana	428	Soft—free burning	Greene	13,099
Indiana	435	Soft—caking	Pike	13,545
Indiana	464	Soft briquettes	Parke	11,930
Indian Territory	437	Soft—free burning		13,932
Indian Territory	449	Semi-anthracite		14,682
Kansas	311	Soft—free burning	Linn	12,343
Kentucky	434	Soft—free burning	Union	14,026

‡ Harding and Willard

Table 28-44. Heat Values of Bituminous Coals*—*Continued*

From selected free-burning and caking soft fuels taken from *Bulletin No. 332,* U. S. Geological Survey and *Bulletin No. 23,* U. S. Bureau of Mines

State	Test No.	Kind of fuel	County	B.t.u. per lb. dry coal
Maryland	490	Soft—free burning	Allegany	14,515
Maryland	518	Soft briquettes	Allegany	14,717
Missouri	319	Soft—caking	Randolph	11,747
Montana	477	Lignite—free burning	Carbon	11,628
New Mexico	392	Soft—caking	Colfax	13,059
New Mexico	387	Soft—free burning	Colfax	12,721
Ohio	483	Soft—free burning	Belmont	13,381
Pennsylvania	473	Soft—caking	Indiana	14,240
Pennsylvania	499	Soft—free burning	Cambria	14,119
Pennsylvania	514	Soft briquettes	Westmoreland	14,382
Tennessee	409	Soft briquettes	Claiborne	14,092
Tennessee	368	Soft—free burning	Campbell	14,008
Tennessee	363	Soft—caking	Grundy	13,257
Texas	291	Lignite—free burning	Wood	11,131
Utah	404	Soft—free burning	Summit	12,586
Virginia	482	Anthracite—free burning	Montgomery	12,679
Virginia	507	Soft—caking	Tazewell	14,177
Washington	290	Subbit—free burning	King	11,772
Washington	359	Soft—free burning	Kittitas	12,996
West Virginia	305	Soft—free burning	Marion	13,964
West Virginia	439	Soft—caking	Kanawha	13,995
Wyoming	399	Soft—free burning	Carbon	12,222
Wyoming	400	Subbit—free burning	Unita	12,488

NOTE—These values were obtained at the St. Louis Testing Plant from 139 samples of coal. The heating values of the various coals were established by "actually burning one gram of the air-dried coal in oxygen in a Mahler-bomb calorimeter." These values in B.t.u. give the theoretical maximum thermal value of soft coals

*Harding & Willard

Table 28-45. Names and Sizes of Anthracite or Hard Coal †

Names of sizes	Will pass through	Will not pass through
Buckwheat No. 1 / " No. 2	½-in. mesh	¼-in. mesh
or Rice	¼-in. mesh	⅛-in. mesh
Pea	¾-in. mesh	½-in. mesh
Chestnut, or Nut	1¼-in. mesh	¾-in. mesh
Stove or Range	1¾-in. mesh	1¼-in. mesh
Egg—in the East	2½-in. mesh	1¾-in. mesh
Large Egg—Chicago	4 -in. mesh	2¾-in. mesh
Small Egg—Chicago	2¾-in. mesh	2 -in. mesh
Broken, or Grate	4 -in. mesh	2½-in. mesh

† Harding and Willard

Table 28-46. Calorific Value of Coal

Where a complete analysis of the coal is not obtainable the following formula may be used: B.t.u. per lb. $= 144\,[100 - (w + a)] - 10.8\,wc$, where w and a are the percentages of water and ash, and c is a constant varying with the amount of water. When $w < 3\%$, $c = 4$; when w is between 3 and 4.5%, $c = 6$; w bet. 4.5 and 8.5%, $c = 12$; w bet. 8.5 and 12%, $c = 10$; w bet. 12 and 20%, $c = 8$; w bet. 20 and 28%, $c = 6$; $w > 28\%$, $c = 4$. Also, when C and C_1 are the percentages of fixed and volatile carbon, respectively, and H the percentage of hydrogen, B.t.u. per lb. $= (14{,}600\,C + 20{,}390\,C_1 + 62{,}000\,H) + 100$

Table 28-47. Composition and Heat Values of Anthracite Coal *

Locality	Fixed carbon	Volatile	Moisture	Ash	Sulphur	B.t.u. per lb. of dry coal
Anthracite						
Pennsylvania	78.60	....	....	14.80	0.40	.....
Buckwheat	81.32	3.84	3.88	10.96	0.67	12,200
Wilkes-Barre	76.94	6.42	1.34	15.30	....	11,801
Scranton	79.23	3.73	3.33	13.70	....	12,149
Scranton	84.46	5.37	0.97	9.20	....	12,294
Cross Creek	89.19	1.96	3.62	5.23	....	13,723
Lehigh Valley	75.20	7.36	1.44	16.00	....	12,423
Lykens Valley	76.94	6.21	....	....	....	15,300
Lykens Valley	81.00	5.00	....	....	....	15,300
Wharton	86.40	3.08	3.71	6.22	0.58	15,000
Buck Mt	82.66	3.95	3.04	9.88	0.46	15,070
Beaver Meadow	88.94	2.38	1.50	7.11	0.01	.....
Lackawanna	87.74	3.91	2.12	6.35	0.12	.....
Rhode Island	85.00	....	....	7.00	0.90	.....
Arkansas	74.49	14.73	1.52	9.26	....	13,217
Semi-Anthracite						
Pennsylvania, Loyalsock	83.34	8.10	1.30	6.23	1.03	15,400
Bernice	82.52	3.56	0.96	3.27	0.24	15,050
Bernice	89.39	8.56	0.97	9.34	1.04	15,475
Wilkes-Barre	88.90	7.68	....	3.49	....	14,199
Lycoming Creek	71.53	13.84	0.67	13.96	0.03	.....
Virginia, Natural Coke	75.08	12.44	1.12	11.38	0.47	.....
Arkansas	74.06	14.93	1.35	9.66	....	.....
Indian Territory	73.21	13.65	5.11	8.03	1.18	13,662
Maryland, Easby	83.60	16.40	....	....	....	11,207

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Table 28-48. Weight of Materials

Dry woods

Material	Weight in lb. of one cu. ft.	Material	Weight in lb. of one cu. ft.	Material	Weight in lb. of one cu. ft.
Ash	43–53	Fir, Spruce	30–44	Oak—American red	54
Beech	43–53	Greenheart	70	" English	48–58
Birch	40–46	Hornbeam	47	Pine—red	30–44
Boxwood	57–83	Larch	31–37	" white	27–34
Cork	15	Lignum-vitae	83	" yellow	29–41
Ebony	70–83	Mahogany—Honduras	35	Teak	41–55
Elm	34–45	" Spanish	53		

Stones, earth, etc.

Material	Weight in lb. of one cu. ft.	Material	Weight in lb. of one cu. ft.	Material	Weight in lb. of one cu. ft.
Asphaltum	64–112	Glass—flint	187	Mud—dry and close	80–110
Brick—common	100–125	" plate	169	" wet and fluid	104–120
" fire	137–150	Granite	164–175	Sand—dry	88–110
Cement—Portland	80– 90	Gravel	90–125	" wet	118–129
Clay	120	Grindstone	134	Sandstone	130–170
Concrete	120–140	Lime—quick	52	Victoria stone (crushed granite, Portland cement, silica)	144
Earth	77–120	Limestone and marbles	150–179		
Glass—crown	156	Mortar—hardened	88–118		

Table 28-49. Weight of Materials—*Continued*

Metals and Alloys

Material		Specific gravity	Weight in lb. of one cu. ft.	Weight in lb. of one cu. in.	Cu. in. in one lb.
Aluminum—cast		2.569	160	.093	10.80
" wrought		2.681	167	.097	10.35
" bronze		7.787	485	.281	3.56
Antimony		6.712	418	.242	4.13
Arsenic		5.748	358	.207	4.83
Bismuth		9.827	612	.354	2.82
Brass—cast	from	7.868	490	.284	3.53
	to	8.430	525	.304	3.29
	average	8.109	505	.292	3.42
" Muntz metal		8.221	512	.296	3.37
" naval (rolled)		8.510	530	.307	3.26
" sheet		8.462	527	.305	3.28
" wire		8.558	533	.308	3.24
Bronze (gun-metal)	from	8.478	528	.306	3.27
	to	8.863	552	.319	3.13
	average	8.735	544	.315	3.18
Copper—cast		8.622	537	.311	3.22
" hammered		8.927	556	.322	3.11
" sheet		8.815	549	.318	3.15
" wire		8.895	554	.321	3.12
Gold (pure)		19.316	1203	.696	1.44
" standard 22 carat fine (Gold 11—Copper 1)		17.502	1090	.631	1.59
Iron—cast	from	6.904	430	.249	4.02
	to	7.386	499	.266	3.76
	average	7.209	464	.260	3.85
Iron—wrought	from	7.547	470	.272	3.56
	to	7.803	486	.281	3.68
	average	7.707	480	.278	3.60
Lead—cast		11.368	708	.410	2.44
" sheet		11.432	712	.412	2.43
Manganese		8.012	499	.289	3.46
Nickel—cast		8.285	516	.299	3.35
" rolled		8.687	541	.313	3.19
Platinum		21.516	1340	.775	1.29
Silver		10.517	655	.379	2.64
Steel	from	7.820	487	.282	3.55
	to	7.916	493	.285	3.51
	average	7.868	490	.284	3.53
Tin		7.418	462	.267	3.74
White Metal (Babbitt's)		7.322	456	.264	3.79
Zinc—cast		6.872	428	.248	4.04
" sheet		7.209	449	.260	3.85

Table 28-50. Specific Heat and Densities of Building Materials *

Building materials	Specific heat	Building materials	Specific heat	Densities in lb per cu. ft.	
Brickwork	0.1950	Oakwood	0.5700	Stonework	160
Concrete	0.2700	Birch	.4800	Wood	40
Masonry	.2159	Glass	.1977	Slate	170
Plaster	.2000	Steel	.1165	Plaster	90
Pinewood	.4670				

* Harding and Willard

Table 28-51. Specific Heats of Various Substances †

Solids

	Temperature,* degrees fahrenheit	Specific heat		Temperature,* degrees fahrenheit	Specific heat
Copper	59–460	0.0951	Glass (normal ther. 16^{III})	66–212	0.1988
Gold	32–212	.0316	Lead	59	.0299
Wrought iron	59–212	.1152	Platinum	32–212	.0323
Cast iron	68–212	.1200	Silver	32–212	.0559
Steel (hard)	68–208	.1175	Tin	105–64	.0518
Steel (soft)	68–208	.1165	Ice	......	.5040
Zinc	32–212	.0935	Sulphur (newly fused)	......	.2025
Brass (yellow)	32	.0883			

Liquids

	Temperature,* degrees fahrenheit	Specific heat		Temperature,* degrees fahrenheit	Specific heat
Water	59	1.0000	Sulphur (melted)	246–297	0.2350
Alcohol	{32 176	0.5475 .7694	Tin (melted)	......	.637
Mercury	32	.3346	Sea-water (sp. gr. 1.0043)	64	.980
Benzol	{50 122	.4066 .4502	Sea-water (sp. gr. 1.0463)	64	.903
Glycerine	59–102	......	Oil of turpentine	32	.411
Lead (melted)	to 360	.0410	Petroleum	64–210	.498
			Sulphuric acid	68–133	.3363
			Olive oil	......	.309

Gases

	Temperature,* degrees fahrenheit	Specific heat at constant pressure	Specific heat at constant volume		Temperature,* degrees fahrenheit	Specific heat at constant pressure	Specific heat at constant volume
Air	32–392	0.2375	0.1693	Carbon monoxide	41–208	0.2425	0.1728
Oxygen	55–405	.2175	.1553	Carbon dioxide	52–417	.2169	.1535
Nitrogen	32–392	.2438	.1729	Methane	64–406	.5929	.4505
Hydrogen	54–388	3.4090	2.4141	Blast-fur. gas (approx.)		.2277	......
				Flue gas (approx.)	......	.2400	......

* When one temperature alone is given the "true" specific heat is given; otherwise the value is the "mean" specific heat for the range of temperature given

† Harding and Willard

Table 28-52. Tensile Strength of Materials

Average value in pounds per square inch

				Woods	
Antimony	1053	Gold	20384	Ash	11000 to 17000
Aluminum—castings	15000	Iron—cast	25000	Beech	11500 to 18000
" sheet	24000	" "	18000	Cedar	10300 to 11400
" bars	28000	" wrought	45000	Chestnut	10500
Brass—yellow	26880	Lead—cast	1800	Elm	13000 to 13489
Bronze—cast	34000	" rolled sheet	3320	Hemlock	8700
" delta metal—cast	44800	Platinum wire	53000	Hickory	12800 to 18000
" " " rolled	67200	Puddled semi-steel	35000 to 42000	Locust	20500 to 24800
" gun metal	32000	Silver—cast	40000	Maple	10500 to 10584
" phosphor	40000	Steel—cast	60000 to 80000	Oak—white	10253 to 19500
" manganese	62720	" forgings	60000 to 95000	Pine—white	10000 to 12000
" Tobin	78500	Tin—cast	3360	" yellow	12600 to 19200
Copper—cast	22400	Zinc—cast	3360	Spruce	10000 to 19500
" sheet	30240	" sheet	15680	Walnut, black	9286 to 16000
" wire	40000				
Cast Steel	80000				

Table 28-53. Lineal Expansion of Solids at Ordinary Temperatures

(Tabular values represent increase per foot per 100-deg. increase in temperature, fahr. or cent.)

Substance	Temperature conditions* deg. fahr.	Coefficient per 100 deg. fahr.	Coefficient per 100 deg. cent.
Brass (cast)	32 to 212	.001042	.001875
Brass (wire)	32 to 212	.001072	.001930
Copper	32 to 212	.000926	.001666
Glass (English flint)	32 to 212	.000451	.000812
Glass (French flint)	32 to 212	.000484	.000872
Gold	32 to 212	.000816	.001470
Granite (average)	32 to 212	.000482	.000868
Iron (cast)	104	.000589	.001061
Iron (soft forged)	0 to 212	.000634	.001141
Iron (wire)	32 to 212	.000800	.001440
Lead	32 to 212	.001505	.002709
Mercury	32 to 212	.009984†	.017971†
Platinum	104	.000499	.000899
Limestone	32 to 212	.000139	.000251
Silver	104	.001067	.001921
Steel (Bessemer rolled, hard)	0 to 212	.00056	.00101
Steel (Bessemer rolled, soft)	0 to 212	.00063	.00117
Steel (cast, French)	104	.000734	.001322
Steel (cast annealed, English)	104	.000608	.001095

* Where range of temperature is given, coefficient is mean over range

† Coefficient of cubical expansion

Table 28-54. Decimal Equivalents of Fractions of an Inch

Fractions				Decimals	Fractions				Decimals	Fractions				Decimals
$\frac{1}{64}$	..	..	..	.015625	$\frac{23}{64}$	..	..	..	.359375	$\frac{45}{64}$	..	..	..	.703125
..	$\frac{1}{32}$	..	..	.03125	..	..	..	$\frac{3}{8}$	.375	..	$\frac{23}{32}$	..	..	.71875
$\frac{3}{64}$	..	..	..	.046875	$\frac{25}{64}$	..	..	..	.390625	$\frac{47}{64}$	..	..	..	.734375
..	..	$\frac{1}{16}$	..	.0625	..	$\frac{13}{32}$	..	..	.40625	..	..	..	$\frac{3}{4}$	.75
$\frac{5}{64}$	..	..	..	.078125	$\frac{27}{64}$	..	..	..	.421875	$\frac{49}{64}$	..	..	..	.765625
..	$\frac{3}{32}$	..	..	.09375	..	..	$\frac{7}{16}$	..	.4375	..	$\frac{25}{32}$	..	..	.78125
$\frac{7}{64}$	..	..	..	.109375	$\frac{29}{64}$	..	..	..	.453125	$\frac{51}{64}$	..	..	..	.796875
..	..	..	$\frac{1}{8}$	.125	..	$\frac{15}{32}$	..	..	.46875	..	..	$\frac{13}{16}$	..	.8125
$\frac{9}{64}$	..	..	..	.140625	$\frac{31}{64}$	..	..	..	.484375	$\frac{53}{64}$	..	..	..	.828125
..	$\frac{5}{32}$	..	..	.15625	..	..	..	$\frac{1}{2}$	.5	..	$\frac{27}{32}$	..	..	.84375
$\frac{11}{64}$	..	..	..	.171875	$\frac{33}{64}$	..	..	..	.515625	$\frac{55}{64}$	..	..	..	.859375
..	..	$\frac{3}{16}$	..	.1875	..	$\frac{17}{32}$	..	..	.53125	..	..	..	$\frac{7}{8}$	.875
$\frac{13}{64}$	..	..	..	.203125	$\frac{35}{64}$	..	..	..	.546875	$\frac{57}{64}$	..	..	..	.890625
..	$\frac{7}{32}$	..	..	.21875	..	..	$\frac{9}{16}$	..	.5625	..	$\frac{29}{32}$	..	..	.90625
$\frac{15}{64}$	..	..	..	.234375	$\frac{37}{64}$	..	..	..	.578125	$\frac{59}{64}$	..	..	..	.921875
..	..	..	$\frac{1}{4}$	.25	..	$\frac{19}{32}$	..	..	.59375	..	..	$\frac{15}{16}$	..	.9375
$\frac{17}{64}$	..	..	..	.265625	$\frac{39}{64}$	..	..	..	.609375	$\frac{61}{64}$	..	..	..	.953125
..	$\frac{9}{32}$	..	..	.28125	..	..	..	$\frac{5}{8}$	.625	..	$\frac{31}{32}$	..	..	.96875
$\frac{19}{64}$	..	..	..	.296875	$\frac{41}{64}$	..	..	..	.640625	$\frac{63}{64}$	..	..	..	.984375
..	..	$\frac{5}{16}$	..	.3125	..	$\frac{21}{32}$	..	..	.65625	..	..	..	1	1.00
$\frac{21}{64}$	..	..	..	.328125	$\frac{43}{64}$	..	..	..	.671875					
..	$\frac{11}{32}$	..	..	.34375	..	..	$\frac{11}{16}$	..	.6875					

Table 28-55. Decimals of a Foot for Inches and Fractions of an Inch

Inch	0″	1″	2″	3″	4″	5″	6″	7″	8″	9″	10″	11″
0	0	.0833	.1667	.2500	.3333	.4167	.5000	.5833	.6667	.7500	.8333	.9167
1/32	.0026	.0859	.1693	.2526	.3359	.4193	.5026	.5859	.6693	.7526	.8359	.9193
1/16	.0052	.0885	.1719	.2552	.3385	.4219	.5052	.5885	.6719	.7552	.8385	.9219
3/32	.0078	.0911	.1745	.2578	.3411	.4245	.5078	.5911	.6745	.7578	.8411	.9245
1/8	.0104	.0937	.1771	.2604	.3437	.4271	.5104	.5937	.6771	.7604	.8437	.9271
5/32	.0130	.0964	.1797	.2630	.3464	.4297	.5130	.5964	.6797	.7630	.8464	.9297
3/16	.0156	.0990	.1823	.2656	.3490	.4323	.5156	.5990	.6823	.7656	.8490	.9323
7/32	.0182	.1016	.1849	.2682	.3516	.4349	.5182	.6016	.6849	.7682	.8516	.9349
1/4	.0208	.1042	.1875	.2708	.3542	.4375	.5208	.6042	.6875	.7708	.8542	.9375
9/32	.0234	.1068	.1901	.2734	.3568	.4401	.5234	.6068	.6901	.7734	.8568	.9401
5/16	.0260	.1094	.1927	.2760	.3594	.4427	.5260	.6094	.6927	.7760	.8594	.9427
11/32	.0286	.1120	.1953	.2786	.3620	.4453	.5286	.6120	.6953	.7786	.8620	.9453
3/8	.0312	.1146	.1979	.2812	.3646	.4479	.5312	.6146	.6979	.7812	.8646	.9479
13/32	.0339	.1172	.2005	.2839	.3672	.4505	.5339	.6172	.7005	.7839	.8672	.9505
7/16	.0365	.1198	.2031	.2865	.3698	.4531	.5365	.6198	.7031	.7865	.8698	.9531
15/32	.0391	.1224	.2057	.2891	.3724	.4557	.5391	.6224	.7057	.7891	.8724	.9557
1/2	.0417	.1250	.2083	.2917	.3750	.4583	.5417	.6250	.7083	.7917	.8750	.9583
17/32	.0443	.1276	.2109	.2943	.3776	.4609	.5443	.6276	.7109	.7943	.8776	.9609
9/16	.0469	.1302	.2135	.2969	.3802	.4635	.5469	.6302	.7135	.7969	.8802	.9635
19/32	.0495	.1328	.2161	.2995	.3828	.4661	.5495	.6328	.7161	.7995	.8828	.9661
5/8	.0521	.1354	.2188	.3021	.3854	.4688	.5521	.6354	.7188	.8021	.8854	.9688
21/32	.0547	.1380	.2214	.3047	.3880	.4714	.5547	.6380	.7214	.8047	.8880	.9714
11/16	.0573	.1406	.2240	.3073	.3906	.4740	.5573	.6406	.7240	.8073	.8906	.9740
23/32	.0599	.1432	.2266	.3099	.3932	.4766	.5599	.6432	.7266	.8099	.8932	.9766
3/4	.0625	.1458	.2292	.3125	.3958	.4792	.5625	.6458	.7292	.8125	.8958	.9792
25/32	.0651	.1484	.2318	.3151	.3984	.4818	.5651	.6484	.7318	.8151	.8984	.9818
13/16	.0677	.1510	.2344	.3177	.4010	.4844	.5677	.6510	.7344	.8177	.9010	.9844
27/32	.0703	.1536	.2370	.3203	.4036	.4870	.5703	.6536	.7370	.8203	.9036	.9870
7/8	.0729	.1562	.2396	.3229	.4062	.4896	.5729	.6562	.7396	.8229	.9062	.9896
29/32	.0755	.1589	.2422	.3255	.4089	.4922	.5755	.6589	.7422	.8255	.9089	.9922
15/16	.0781	.1615	.2448	.3281	.4115	.4948	.5781	.6615	.7448	.8281	.9115	.9948
31/32	.0807	.1641	.2474	.3307	.4141	.4974	.5807	.6641	.7474	.8307	.9141	.9974
1	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1.0000

Table 28-56. Decimals of a Foot Equivalent to Inches and Fractions of an Inch

Inches	0″	1/8″	1/4″	3/8″	1/2″	5/8″	3/4″	7/8″
0	0	.01042	.02083	.03125	.04166	.05208	.06250	.07292
1	.0833	.0937	.1042	.1146	.1250	.1354	.1459	.1563
2	.1667	.1771	.1875	.1979	.2083	.2188	.2292	.2396
3	.2500	.2604	.2708	.2813	.2917	.3021	.3125	.3229
4	.3333	.3437	.3542	.3646	.3750	3854	.3958	.4063
5	.4167	.4271	.4375	.4479	.4583	.4688	.4792	.4896
6	.5000	.5104	.5208	.5313	.5417	.5521	.5625	.5729
7	.5833	.5937	.6042	.6146	.6250	.6354	.6459	.6563
8	.6667	.6771	.6875	.6979	.7083	.7188	.7292	.7396
9	.7500	.7604	.7708	.7813	.7917	.8021	.8125	.8229
10	.8333	.8437	.8542	.8646	.8750	.8854	.8958	.9063
11	.9167	.9271	.9375	.9479	.9583	.9688	.9792	.9896

Table 28-57. Circumferences and Areas of Circles Advancing by Eighths

Diam.	Circum.	Area	Diam.	Circum.	Area	Diam.	Circum.	Area	Diam.	Circum.	Area
1/64	.04909	.00019	2. 11/16	8.4430	5.6727	7.	21.991	38.485	14. 1/4	44.768	159.48
1/32	.09818	.00077	3/4	8.6394	5.9396	1/8	22.384	39.871	3/8	45.160	162.30
3/64	.14726	.00173	13/16	8.8357	6.2126	1/4	22.776	41.282	1/2	45.553	165.13
1/16	.19635	.00307	7/8	9.0321	6.4918	3/8	23.169	42.718	5/8	45.946	167.99
3/32	.29452	.00690	15/16	9.2284	6.7771	1/2	23.562	44.179	3/4	46.338	170.87
1/8	.39270	.01227				5/8	23.955	45.664	7/8	46.731	173.78
5/32	.49087	.01917	3.	9.4248	7.0686	3/4	24.347	47.173			
3/16	.58905	.02761	1/16	9.6211	7.3662	7/8	24.740	48.707	15.	47.124	176.71
7/32	.68722	.03758	1/8	9.8175	7.6699				1/8	47.517	179.67
			3/16	10.014	7.9798	8.	25.133	50.265	1/4	47.909	182.65
1/4	.78540	.04909	1/4	10.210	8.2958	1/8	25.525	51.849	3/8	48.302	185.66
9/32	.88357	.06213	5/16	10.407	8.6179	1/4	25.918	53.456	1/2	48.695	188.69
5/16	.98175	.07670	3/8	10.603	8.9462	3/8	26.311	55.088	5/8	49.087	191.75
11/32	1.0799	.09281	7/16	10.799	9.2806	1/2	26.704	56.745	3/4	49.480	194.83
3/8	1.1781	.11045	1/2	10.996	9.6211	5/8	27.096	58.426	7/8	49.873	197.93
13/32	1.2763	.12962	9/16	11.192	9.9678	3/4	27.489	60.132			
7/16	1.3744	.15033	5/8	11.388	10.321	7/8	27.882	61.862	16.	50.265	201.06
15/32	1.4726	.17257	11/16	11.585	10.680				1/8	50.658	204.22
			3/4	11.781	11.045	9.	28.274	63.617	1/4	51.051	207.39
1/2	1.5708	.19635	13/16	11.977	11.416	1/8	28.667	65.397	3/8	51.444	210.60
17/32	1.6690	.22166	7/8	12.174	11.793	1/4	29.060	67.201	1/2	51.836	213.82
9/16	1.7671	.24850	15/16	12.370	12.177	3/8	29.452	69.029	5/8	52.229	217.08
19/32	1.8653	.27688				1/2	29.845	70.882	3/4	52.622	220.35
5/8	1.9635	.30680	4.	12.566	12.566	5/8	30.238	72.760	7/8	53.014	223.65
21/32	2.0617	.33824	1/16	12.763	12.962	3/4	30.631	74.662			
11/16	2.1598	.37122	1/8	12.959	13.364	7/8	31.023	76.589	17.	53.407	226.98
23/32	2.2580	.40574	3/16	13.155	13.772				1/8	53.800	230.33
			1/4	13.352	14.186	10.	31.416	78.540	1/4	54.192	233.71
3/4	2.3562	.44179	5/16	13.548	14.607	1/8	31.809	80.516	3/8	54.585	237.10
25/32	2.4544	.47937	3/8	13.744	15.033	1/4	32.201	82.516	1/2	54.978	240.53
13/16	2.5525	.51849	7/16	13.941	15.466	3/8	32.594	84.541	5/8	55.371	243.98
27/32	2.6507	.55914	1/2	14.137	15.904	1/2	32.987	86.590	3/4	55.763	247.45
7/8	2.7489	.60132	9/16	14.334	16.349	5/8	33.379	88.664	7/8	56.156	250.95
29/32	2.8471	.64504	5/8	14.530	16.800	3/4	33.772	90.763			
15/16	2.9452	.69029	11/16	14.726	17.257	7/8	34.165	92.886	18.	56.549	254.47
31/32	3.0434	.73708	3/4	14.923	17.721				1/8	56.941	258.02
			13/16	15.119	18.190	11.	34.558	95.033	1/4	57.334	261.59
1.	3.1416	.7854	7/8	15.315	18.665	1/8	34.950	97.205	3/8	57.727	265.18
1/16	3.3379	.8866	15/16	15.512	19.147	1/4	35.343	99.402	1/2	58.119	268.80
1/8	3.5343	.9940				3/8	35.736	101.62	5/8	58.512	272.45
3/16	3.7306	1.1075	5.	15.708	19.635	1/2	36.128	103.87	3/4	58.905	276.12
1/4	3.9270	1.2272	1/16	15.904	20.129	5/8	36.521	106.14	7/8	59.298	279.81
5/16	4.1233	1.3530	1/8	16.101	20.629	3/4	36.914	108.43			
3/8	4.3197	1.4849	3/16	16.297	21.135	7/8	37.306	110.75	19.	59.690	283.53
7/16	4.5160	1.6230	1/4	16.493	21.648				1/8	60.083	287.27
1/2	4.7124	1.7671	5/16	16.690	22.166	12.	37.699	113.10	1/4	60.476	291.04
9/16	4.9087	1.9175	3/8	16.886	22.691	1/8	38.092	115.47	3/8	60.868	294.83
5/8	5.1051	2.0739	7/16	17.082	23.221	1/4	38.485	117.86	1/2	61.261	298.65
11/16	5.3014	2.2365	1/2	17.279	23.758	3/8	38.877	120.28	5/8	61.654	302.49
3/4	5.4978	2.4053	9/16	17.475	24.301	1/2	39.270	122.72	3/4	62.046	306.35
13/16	5.6941	2.5802	5/8	17.671	24.850	5/8	39.663	125.19	7/8	62.439	310.24
7/8	5.8905	2.7612	11/16	17.868	25.406	3/4	40.055	127.68			
15/16	6.0868	2.9483	3/4	18.064	25.967	7/8	40.448	130.19	20.	62.832	314.16
			13/16	18.261	26.535				1/8	63.225	318.10
2.	6.2832	3.1416	7/8	18.457	27.109	13.	40.841	132.73	1/4	63.617	322.06
1/16	6.4795	3.3410	15/16	18.653	27.688	1/8	41.233	135.30	3/8	64.010	326.05
1/8	6.6759	3.5466				1/4	41.626	137.89	1/2	64.403	330.06
3/16	6.8722	3.7583	6.	18.850	28.274	3/8	42.019	140.50	5/8	64.795	334.10
1/4	7.0686	3.9761	1/8	19.242	29.465	1/2	42.412	143.14	3/4	65.188	338.16
5/16	7.2649	4.2000	1/4	19.635	30.680	5/8	42.804	145.80	7/8	65.581	342.25
3/8	7.4613	4.4301	3/8	20.028	31.919	3/4	43.197	148.49			
7/16	7.6576	4.6664	1/2	20.420	33.183	7/8	43.590	151.20	21.	65.973	346.36
1/2	7.8540	4.9087	5/8	20.813	34.472				1/8	66.366	350.50
9/16	8.0503	5.1572	3/4	21.206	35.785	14.	43.982	153.94	1/4	66.759	354.66
5/8	8.2467	5.4119	7/8	21.598	37.122	1/8	44.375	156.70	3/8	67.152	358.84

Table 28-57. Circumferences and Areas of Circles Advancing by Eighths—*Continued*

Diam.	Circum.	Area
21.½	67.544	363.05
⅝	67.937	367.28
¾	68.330	371.54
⅞	68.722	375.83
.	69.115	380.13
⅛	69.508	384.46
¼	69.900	388.82
⅜	70.293	393.20
½	70.686	397.61
⅝	71.079	402.04
¾	71.471	406.49
⅞	71.864	410.97
23.	72.257	415.48
⅛	72.649	420.00
¼	73.042	424.56
⅜	73.435	429.13
½	73.827	433.74
⅝	74.220	438.36
¾	74.613	443.01
⅞	75.006	447.69
24.	75.398	452.39
⅛	75.791	457.11
¼	76.184	461.86
⅜	76.576	466.64
½	76.969	471.44
⅝	77.362	476.26
¾	77.754	481.11
⅞	78.147	485.98
25.	78.540	490.87
⅛	78.933	495.79
¼	79.325	500.74
⅜	79.718	505.71
½	80.111	510.71
⅝	80.503	515.72
¾	80.896	520.77
⅞	81.289	525.84
26.	81.681	530.93
⅛	82.074	536.05
¼	82.467	541.19
⅜	82.860	546.35
½	83.252	551.55
⅝	83.645	556.76
¾	84.038	562.00
⅞	84.430	567.27
27.	84.823	572.56
⅛	85.216	577.87
¼	85.608	583.21
⅜	86.001	588.57
½	86.394	593.96
⅝	86.786	599.37
¾	87.179	604.81
⅞	87.572	610.27
28.	87.965	615.75
⅛	88.357	621.26
¼	88.750	626.80
⅜	89.143	632.36
½	89.535	637.94
⅝	89.928	643.55

Diam.	Circum.	Area
28.¾	90.321	649.18
⅞	90.713	654.84
29.	91.106	660.52
⅛	91.499	666.23
¼	91.892	671.96
⅜	92.284	677.71
½	92.677	683.49
⅝	93.070	689.30
¾	93.462	695.13
⅞	93.855	700.98
30.	94.248	706.86
⅛	94.640	712.76
¼	95.033	718.69
⅜	95.426	724.64
½	95.819	730.62
⅝	96.211	736.62
¾	96.604	742.64
⅞	96.997	748.69
31.	97.389	754.77
⅛	97.782	760.87
¼	98.175	766.99
⅜	98.567	773.14
½	98.960	779.31
⅝	99.353	785.51
¾	99.746	791.73
⅞	100.138	797.98
32.	100.531	804.25
⅛	100.924	810.54
¼	101.316	816.86
⅜	101.709	823.21
½	102.102	829.58
⅝	102.494	835.97
¾	102.887	842.39
⅞	103.280	848.83
33.	103.673	855.30
⅛	104.065	861.79
¼	104.458	868.31
⅜	104.851	874.85
½	105.243	881.41
⅝	105.636	888.00
¾	106.029	894.62
⅞	106.421	901.26
34.	106.814	907.92
⅛	107.207	914.61
¼	107.600	921.32
⅜	107.992	928.06
½	108.385	934.82
⅝	108.778	941.61
¾	109.170	948.42
⅞	109.563	955.25
35.	109.956	962.11
⅛	110.348	969.00
¼	110.741	975.91
⅜	111.134	982.84
½	111.527	989.80
⅝	111.919	996.78
¾	112.312	1000.38
⅞	112.705	1010.8

Diam.	Circum.	Area
36.	113.097	1017.9
⅛	113.490	1025.0
¼	113.883	1032.1
⅜	114.275	1039.2
½	114.668	1046.3
⅝	115.061	1053.5
¾	115.454	1060.7
⅞	115.846	1068.0
37.	116.239	1075.2
⅛	116.632	1082.5
¼	117.024	1089.8
⅜	117.417	1097.1
½	117.810	1104.5
⅝	118.202	1111.8
¾	118.596	1119.2
⅞	118.988	1126.7
38.	119.381	1134.1
⅛	119.773	1141.2
¼	120.166	1149.2
⅜	120.559	1156.6
½	120.951	1164.2
⅝	121.344	1171.7
¾	121.737	1179.3
⅞	122.129	1186.9
39.	122.522	1194.6
⅛	122.915	1202.3
¼	123.308	1210.0
⅜	123.700	1217.7
½	124.093	1225.4
⅝	124.486	1233.2
¾	124.878	1241.0
⅞	125.271	1248.8
40.	125.664	1256.6
⅛	126.056	1264.5
¼	126.449	1272.4
⅜	126.842	1280.3
½	127.235	1288.2
⅝	127.627	1296.2
¾	128.020	1304.2
⅞	128.413	1312.2
41.	128.805	1320.3
⅛	129.198	1328.3
¼	129.591	1336.4
⅜	129.983	1344.5
½	130.376	1352.7
⅝	130.769	1360.8
¾	131.161	1369.0
⅞	131.554	1377.2
42.	131.947	1385.4
⅛	132.340	1393.7
¼	132.732	1402.0
⅜	133.125	1410.3
½	133.518	1418.6
⅝	133.910	1427.0
¾	134.303	1435.4
⅞	134.696	1443.8
43.	135.088	1452.2
⅛	135.481	1460.7

Diam.	Circum.	Area
43.¼	135.874	1469.1
⅜	136.267	1477.6
½	136.659	1486.2
⅝	137.052	1494.7
¾	137.445	1503.3
⅞	137.837	1511.9
44.	138.230	1520.5
⅛	138.623	1529.2
¼	139.015	1537.9
⅜	139.408	1546.6
½	139.801	1555.3
⅝	140.194	1564.0
¾	140.586	1572.8
⅞	140.979	1581.6
45.	141.372	1590.4
⅛	141.764	1599.3
¼	142.157	1608.2
⅜	142.550	1617.0
½	142.942	1626.0
⅝	143.335	1634.9
¾	143.728	1643.9
⅞	144.121	1652.9
46.	144.513	1661.9
⅛	144.906	1670.9
¼	145.299	1680.0
⅜	145.691	1689.1
½	146.084	1698.2
⅝	146.477	1707.4
¾	146.869	1716.5
⅞	147.262	1725.7
47.	147.655	1734.9
⅛	148.048	1744.2
¼	148.440	1753.5
⅜	148.833	1762.7
½	149.226	1772.1
⅝	149.618	1781.4
¾	150.011	1790.8
⅞	150.404	1800.1
48.	150.796	1809.6
⅛	151.189	1819.0
¼	151.582	1828.5
⅜	151.975	1837.9
½	152.367	1847.5
⅝	152.760	1857.0
¾	153.153	1866.5
⅞	153.545	1876.1
49.	153.938	1885.7
⅛	154.331	1895.4
¼	154.723	1905.0
⅜	155.116	1914.7
½	155.509	1924.4
⅝	155.902	1934.2
¾	156.294	1943.9
⅞	156.687	1953.7
50.	157.080	1963.5

Table 28-58. Fractional Equivalents, Powers and Roots of Numbers

Number	Frac. equiv.	Square root	Cube root	Square	Cube	$\sqrt{2G \times No.}$	Number	Frac. equiv.	Square root	Cube root	Square	Cube	$\sqrt{2G \times No.}$
.01	..	.1	.2154	.0001	.000001	.802	.3281	$\frac{21}{64}$	.5728	.6897	.1077	.03533	4.594
.0156	$\frac{1}{64}$	.125	.25	.0002441	.000003815	1.003	.33	..	.5745	.6910	.1089	.03594	4.607
.02	..	.1414	.2714	.0004	.000008	1.134	.34	..	.5831	.6980	.1156	.03930	4.677
.03	..	.1732	.3107	.0009	.000027	1.389	.3438	$\frac{11}{32}$	.5863	.7005	.1182	.04062	4.702
.0313	$\frac{1}{32}$	.1768	.3150	.0009766	.00003052	1.418	.35	..	.5916	.7047	.1225	.04288	4.745
.04	..	.2	.3420	.0016	.000064	1.604	.3594	$\frac{23}{64}$	.5995	.7110	.1292	.04641	4.808
.0469	$\frac{3}{64}$	.2165	.3606	.002197	.000103	1.756	.36	..	.6	.7114	.1296	.04666	4.812
.05	..	.2236	.3684	.0025	.000125	1.793	.37	..	.6083	.7179	.1369	.05065	4.879
.06	..	.2449	.3915	.0036	.000216	1.965	.375	$\frac{3}{8}$	.6124	.7211	.1406	.05273	4.911
.0625	$\frac{1}{16}$	.25	.3968	.003906	.0002441	2.005	.38	..	.6164	.7243	.1444	.05487	4.944
.07	..	.2646	.4121	.0049	.000343	2.122	.39	..	.6245	.7306	.1521	.05932	5.009
.0781	$\frac{5}{64}$	.2795	.4275	.006104	.0004768	2.242	.3906	$\frac{25}{64}$	.625	.7310	.1526	.05960	5.013
.08	..	.2828	.4309	.0064	.000512	2.269	.4	..	.6325	.7368	.16	.64	5.072
.09	..	.3	.4481	.0081	.000729	2.406	.4063	$\frac{13}{32}$	.6374	.7406	.1650	.06705	5.112
.0938	$\frac{3}{32}$	.3062	.4543	.008789	.0008240	2.456	.41	..	.6403	.7429	.1681	.06892	5.135
.1	..	.3162	.4642	.01	.001	2.537	.42	..	.6481	.7489	.1764	.07409	5.198
.1094	$\frac{7}{64}$	.3307	.4782	.01196	.001308	2.653	.4219	$\frac{27}{64}$	.6495	.75	.1780	.07508	5.209
.11	..	.3317	.4791	.0121	.001331	2.660	.43	..	.6557	.7548	.1849	.07951	5.259
.12	..	.3464	.4932	.0144	.001728	2.778	.4375	$\frac{7}{16}$	.6614	.7591	.1914	.08374	5.305
.125	$\frac{1}{8}$	.3536	.5	.01562	.001953	2.836	.44	..	.6633	.7606	.1936	.08518	5.320
.13	..	.3606	.5066	.0169	.002197	2.892	.45	..	.6708	.7663	.2025	.09113	5.380
.14	..	.3742	.5193	.0196	.002744	3.001	.4531	$\frac{29}{64}$	.6732	.7681	.2053	.09304	5.399
.1406	$\frac{9}{64}$	.375	.5200	.01978	.002781	3.008	.46	..	.6782	.7719	.2116	.09734	5.440
.15	..	.3873	.5313	.0225	.003375	3.106	.4688	$\frac{15}{32}$	.6847	.7768	.2197	.1030	5.491
.1563	$\frac{5}{32}$	.3953	.5386	.02441	.003815	3.170	.47	..	.6856	.7775	.2209	.1038	5.498
.16	..	.4	.5429	.0256	.004096	3.208	.48	..	.6928	.7830	.2304	.1106	5.557
.17	..	.4123	.5540	.0289	.004913	3.307	.4844	$\frac{31}{64}$	.6960	.7853	.2346	.1136	5.582
.1719	$\frac{11}{64}$	.4146	.5560	.02954	.005077	3.325	.49	..	.7	.7884	.2401	.1176	5.614
.18	..	.4243	.5646	.0324	.005832	3.403	.5	$\frac{1}{2}$	.7071	.7937	.25	.125	5.671
.1875	$\frac{3}{16}$	.433	.5724	.03516	.006592	3.473	.51	..	.7141	.7990	.2601	.1327	5.728
.19	..	.4359	.5749	.0361	.006859	3.496	.5156	$\frac{33}{64}$	.7181	.8019	.2658	.1371	5.759
.20	..	.4472	.5848	.04	.008	3.587	.52	..	.7211	.8042	.2704	.1406	5.784
.2031	$\frac{13}{64}$	.4507	.5878	.04126	.008381	3.615	.53	..	.7280	.8093	.2809	.1489	5.839
.21	..	.4583	.5944	.0441	.009261	3.675	.5313	$\frac{17}{32}$	.7289	.8099	.2822	.1499	5.846
.2188	$\frac{7}{32}$	.4677	.6025	.04785	.01047	3.751	.54	..	.7349	.8143	.2916	.1575	5.894
.22	..	.4690	.6037	.0484	.01065	3.762	.5469	$\frac{35}{64}$	.7395	.8178	.2991	.1636	5.931
.23	..	.4796	.6127	.0529	.01217	3.846	.55	..	.7416	.8193	.3025	.1664	5.948
.2344	$\frac{15}{64}$	.4841	.6165	.05493	.01287	3.883	.56	..	.7483	.8243	.3136	.1756	6.002
.24	..	.4899	.6215	.0576	.01382	3.929	.5625	$\frac{9}{16}$	.75	.8255	.3164	.1780	6.015
.25	$\frac{1}{4}$	.5	.6300	.0625	.01563	4.010	.57	..	.7550	.8291	.3249	.1852	6.055
.26	..	.5099	.6383	.0676	.01758	4.090	.5781	$\frac{37}{64}$	.7603	.8330	.3342	.1932	6.098
.2656	$\frac{17}{64}$	.5154	.6428	.07056	.01874	4.134	.58	..	.7616	.8340	.3364	.1951	6.108
.27	..	.5196	.6463	.0729	.01968	4.167	.59	..	.7681	.8387	.3481	.2054	6.161
.28	..	.5292	.6542	.0784	.02195	4.244	.5938	$\frac{19}{32}$	.7706	.8405	.3525	.2093	6.180
.2813	$\frac{9}{32}$	.5303	.6552	.07910	.02225	4.253	.6	..	.7746	.8434	.36	.2160	6.212
.29	..	.5385	.6619	.0841	.02439	4.319	.6094	$\frac{39}{64}$	.7806	.8478	.3713	.2263	6.261
.2969	$\frac{19}{64}$	.5448	.6671	.08814	.02617	4.370	.61	..	.7810	.8481	.3721	.2270	6.264
.30	..	.5477	.6694	.09	.027	4.393	.62	..	.7874	.8527	.3844	.2383	6.315
.31	..	.5568	.6768	.0961	.02979	4.466	.625	$\frac{5}{8}$	.7906	.8550	.3906	.2441	6.341
.3125	$\frac{5}{16}$	.5590	.6786	.09766	.03052	4.483	.63	..	.7937	.8573	.3969	.2500	6.366
.32	..	.5657	.6840	.1024	.03277	4.537	.64	..	.8	.8618	.4096	.2621	6.416

Table 28-58. Fractional Equivalents, Powers and Roots of Numbers—*Continued*

Number	Frac. equiv.	Square root	Cube root	Square	Cube	$\sqrt{2 G \times No.}$
.6406	41/64	.8004	.8621	.4104	.2629	6.419
.65	..	.8062	.8662	.4225	.2746	6.466
.6563	21/32	.8101	.8690	.4307	.2826	6.497
.66	..	.8124	.8707	.4356	.2875	6.516
.67	..	.8185	.8750	.4489	.3008	6.565
.6719	43/64	.8197	.8759	.4514	.3033	6.574
.68	..	.8246	.8794	.4624	.3144	6.614
.6875	11/16	.8292	.8826	.4727	.3249	6.650
.69	..	.8307	.8837	.4761	.3285	6.662
.70	..	.8367	.8879	.49	.3430	6.710
.7031	45/64	.8395	.8892	.4944	.3476	6.725
.71	..	.8426	.8921	.5041	.3579	6.758
.7188	23/32	.8478	.8958	.5166	.3713	6.799
.72	..	.8485	.8963	.5184	.3732	6.805
.73	..	.8544	.9004	.5329	.3890	6.853
.7344	47/64	.8570	.9022	.5393	.3961	6.873
.74	..	.8602	.9045	.5476	.4052	6.899
.75	3/4	.8660	.9086	.5625	.4219	6.916
.76	..	.8718	.9126	.5776	.4390	6.992
.7656	49/64	.875	.9148	.5862	.4488	7.018
.77	..	.8775	.9166	.5929	.4565	7.038
.78	..	.8832	.9205	.6084	.4746	7.083
.7813	25/32	.8839	.9210	.6104	.4768	7.089
.79	..	.8888	.9244	.6241	.4930	7.129
.7969	51/64	.8927	.9271	.6350	.5060	7.159
.8	..	.8944	.9283	.64	.5120	7.174
.81	..	.9	.9322	.6561	.5314	7.218
.8125	13/16	.9014	.9331	.6602	.5364	7.229
.82	..	.9055	.9360	.6724	.5514	7.263
.8281	53/64	.9100	.9391	.6858	.5679	7.298
.83	..	.9110	.9398	.6889	.5718	7.307
.84	..	.9165	.9435	.7056	.5927	7.351
.8438	27/32	.9186	.9449	.7120	.6007	7.367
.85	..	.9219	.9473	.7225	.6141	7.394
.8594	55/64	.9270	.9507	.7385	.6347	7.435
.86	..	.9274	.9510	.7396	.6361	7.438
.87	..	.9327	.9546	.7569	.6585	7.481
.875	7/8	.9354	.9565	.7656	.6699	7.502
.88	..	.9381	.9583	.7744	.6815	7.524
.89	..	.9434	.9619	.7921	.7050	7.566
.8906	57/64	.9437	.9621	.7932	.7065	7.569
.9	..	.9487	.9655	.81	.7290	7.609
.9063	29/32	.9520	.9677	.8213	.7443	7.635
.91	..	.9539	.9691	.8281	.7536	7.651
.92	..	.9592	.9726	.8464	.7787	7.693
.9219	59/64	.9601	.9732	.8499	.7835	7.701
.93	..	.9644	.9761	.8649	.8044	7.734
.9375	15/16	.9682	.9787	.8789	.8240	7.766
.94	..	.9695	.9796	.8836	.8306	7.776
.95	..	.9747	.9831	.9025	.8574	7.817
.9531	61/64	.9763	.9840	.9084	.8659	7.830

Number	Frac. equiv.	Square root	Cube root	Square	Cube	$\sqrt{2 G \times No.}$
.96	..	.9798	.9865	.9216	.8847	7.858
.9688	31/32	.9843	.9895	.9385	.9091	7.894
.97	..	.9849	.9899	.9409	.9127	7.899
.98	..	.9899	.9933	.9604	.9412	7.940
.9844	63/64	.9922	.9948	.9690	.9538	7.957
.99	..	.9950	.9967	.9801	.9703	7.980
1.	..	1.	1.	1.	1.	8.021
1.1	..	1.049	1.032	1.21	1.331	8.412
1.2	..	1.095	1.063	1.44	1.728	8.786
1.3	..	1.14	1.091	1.69	2.197	9.145
1.4	..	1.183	1.119	1.96	2.744	9.490
1.5	..	1.225	1.1145	2.25	3.375	9.823
1.6	..	1.265	1.170	2.56	4.096	10.14
1.7	..	1.304	1.193	2.89	4.913	10.45
1.8	..	1.342	1.216	3.24	5.832	10.76
1.9	..	1.378	1.239	3.61	6.859	11.06
2.	..	1.414	1.260	4.	8.	11.34
2.1	..	1.449	1.281	4.41	9.261	11.62
2.2	..	1.483	1.301	4.84	10.65	11.90
2.3	..	1.517	1.320	5.29	12.17	12.16
2.4	..	1.549	1.339	5.76	13.82	12.43
2.5	..	1.581	1.357	6.25	15.63	12.68
2.6	..	1.612	1.375	6.76	17.58	12.93
2.7	..	1.643	1.392	7.29	19.68	13.18
2.8	..	1.673	1.409	7.84	21.95	13.42
2.9	..	1.703	1.426	8.41	24.39	13.66
3.	..	1.732	1.442	9.	27.	13.89
3.1	..	1.761	1.458	9.61	29.79	14.12
3.2	..	1.789	1.474	10.24	32.77	14.35
3.3	..	1.817	1.489	10.89	35.94	14.57
3.4	..	1.844	1.504	11.56	39.30	14.79
3.5	..	1.871	1.518	12.25	42.88	15.01
3.6	..	1.897	1.533	12.96	46.66	15.22
3.7	..	1.924	1.547	13.69	50.65	15.43
3.8	..	1.949	1.560	14.44	54.87	15.64
3.9	..	1.975	1.574	15.21	59.32	15.85
4.	..	2.	1.587	16.	64.	16.04
4.1	..	2.025	1.601	16.81	68.92	16.24
4.2	..	2.049	1.613	17.64	74.09	16.44
4.3	..	2.074	1.626	18.49	79.51	16.63
4.4	..	2.098	1.639	19.36	85.18	16.82
4.5	..	2.121	1.651	20.25	91.13	17.01
4.6	..	2.145	1.663	21.16	97.34	17.20
4.7	..	2.168	1.675	22.09	103.8	17.39
4.8	..	2.191	1.687	23.04	110.6	17.57
4.9	..	2.214	1.698	24.01	117.6	17.75
5.	..	2.236	1.710	25.	125.	17.93

Table 28-59. Comparison of Wire Gauges

Thickness in decimals of an inch

Gauge No.	American or Brown & Sharpe's	Birmingham or Stubs	Wash. & Moen	Imperial S. W. G.	London or Old English	United States Standard	Gauge No.	American or Brown & Sharpe's	Birmingham or Stubs	Wash. & Moen	Imperial S. W. G.	London or Old English	United States Standard
0000000			.490	.500		.5	23	.02257	.025	.0258	.024	.027	.028125
000000	.5800		.460	.464		.46875	24	.02010	.022	.0230	.022	.025	.025
00000	.5165		.430	.432		.4375	25	.01790	.020	.0204	.020	.023	.021875
0000	.4600	.454	.3938	.400	.454	.40625	26	.01594	.018	.0181	.018	.0205	.01875
000	.4096	.425	.3625	.372	.425	.375	27	.01420	.016	.0173	.0164	.0187	.0171875
00	.3648	.380	.3310	.348	.380	.34375	28	.01264	.014	.0162	.0148	.0165	.015625
0	.3249	.340	.3065	.324	.340	.3125	29	.01126	.013	.0150	.0136	.0155	.0140625
1	.2893	.300	.2830	.300	.300	.28125	30	.01003	.012	.0140	.0124	.01372	.0125
2	.2576	.284	.2625	.276	.284	.265625	31	.008928	.010	.0132	.0116	.0122	.0109375
3	.2294	.259	.2437	.252	.259	.25	32	.007950	.009	.0128	.0108	.0112	.01015625
4	.2043	.238	.2253	.232	.238	.234375	33	.007080	.008	.0118	.0100	.0102	.009375
5	.1819	.220	.2070	.212	.22	.21875	34	.006305	.007	.0104	.0092	.0095	.00859375
6	.1620	.203	.1920	.192	.203	.203125	35	.005615	.005	.0095	.0084	.0090	.0078125
7	.1443	.180	.1770	.176	.18	.1875	36	.005000	.004	.0090	.0076	.0075	.00703125
8	.1285	.165	.1620	.160	.165	.171875	37	.004453		.0085	.0068	.0065	.006640625
9	.1144	.148	.1483	.144	.148	.15625	38	.003965		.0080	.0060	.0057	.00625
10	.1019	.134	.1350	.128	.134	.140625	39	.003531		.0075	.0052	.0050	
11	.09074	.120	.1205	.116	.12	.125	40	.003145		.0070	.0048	.0045	
12	.08081	.109	.1055	.104	.109	.109375	41	.002800			.0044		
13	.07196	.095	.0915	.092	.095	.09375	42	.002494			.0040		
14	.06408	.083	.0800	.080	.083	.078125	43	.002221			.0036		
15	.05707	.072	.0720	.072	.072	.0703125	44	.001978			.0032		
16	.05082	.065	.0625	.064	.065	.0625	45	.001761			.0028		
17	.04526	.058	.0540	.056	.058	.05625	46	.001568			.0024		
18	.04030	.049	.0475	.048	.049	.05	47	.001397			.0020		
19	.03589	.042	.0410	.040	.040	.04375	48	.001244			.0016		
20	.03196	.035	.0348	.036	.035	.0375	49	.001018			.0012		
21	.02846	.032	.03175	.032	.0315	.034375	50	.0009863			.0010		
22	.02535	.028	.0286	.028	.0295	.03125							

Table 28-60. Useful Factors

1 gal. (U. S.) = 231 cu. in.
= 0.13368 cu. ft.
1 gal (British) = 277.274 cu. in.
1 cu. ft. = 7.4805 gal.
1 cu. ft. water at 60 deg. fahr. = 62.37 lb.
1 gal. water at 60 deg. fahr. = 8.34 lb.
1 cu. ft. water at 212 deg. fahr. = 59.76 lb.
1 gal. water at 212 deg. fahr. = 7.99 lb.
1 barrel water at 60 deg. fahr. = 31½ gal. = 262.7 lb.
1 inch mercury = 1⅛ ft. or 13.6 in. water
= 0.491 lb. per sq. in.
1 lb. per sq. in. pressure = 2.304 ft. water at 60 deg. fahr.
Height of a column of water in feet × 0.434 = lb. pressure per sq. in.
A column of water 1 sq. in. and 2⅓ ft. high = approximately 1 lb.
1 calorie = 3.97 B.t.u.
1 kilogram = 2.2046 lb.
Calories per kilo × 1.8 = B.t.u. per lb.
1 kilowatt (1000 watts) = 1.3405 hp.
1 horsepower = 0.746 kw.
1 kilowatt = 56.9 B.t.u. per min.
= 3414 B.t.u. per hour
1 mech. horsepower = 42.4 B.t.u. per min.
= 2545 B.t.u. per hour
= 33000 ft. lb. per min.
1 boiler horsepower = 33479 B.t.u. per hour
1 B.t.u. = 778 ft. lb.
1 ft. lb. per sec. = 1.356 watts

Table 28-61. Standard Gauges of Sheet Metal

No. of gauge	U. S. Standard				Birmingham or Stubs				No. of gauge
	Thickness, inches		Weight per sq. ft. in lb.		Thickness, inches		Weight per sq. ft. in lb.		
	Fractions	Decimals	Iron	Steel	Fractions	Decimals	Iron	Steel	
7–0	1–2	.5	20.00	20.4	(Approx.)				7–0
6–0	15–32	.46875	18.75	19.125					6–0
5–0	7–16	.4375	17.50	17.85					5–0
4–0	13–32	.40625	16.25	16.575	29–64	.454	18.16	18.52	4–0
3–0	3–8	.375	15.00	15.30	27–64	.425	17.00	17.34	3–0
2–0	11–32	.34375	13.75	14.025	3–8	.38	15.20	15.50	2–0
0	5–16	.3125	12.50	12.75	11–32	.34	13.60	13.87	0
1	9–32	.28125	11.25	11.475	19–64	.3	12.00	12.24	1
2	17–64	.265625	10.625	10.8375	9–32	.284	11.36	11.59	2
3	1–4	.25	10.	10.2	17–64	.259	10.36	10.57	3
4	15–64	.234375	9.375	9.5625	15–64	.238	9.52	9.71	4
5	7–32	.21875	8.75	8.925	7–32	.22	8.80	8.98	5
6	13–64	.203125	8.125	8.2875	13–64	.203	8.12	8.28	6
7	3–16	.1875	7.5	7.65	3–16	.18	7.20	7.34	7
8	11–64	.171875	6.875	7.0125		.165	6.60	6.73	8
					5–32				
9	5–32	.15625	6.25	6.375		.148	5.92	6.04	9
10	9–64	.140625	5.625	5.7375	9–64	.134	5.36	5.47	10
11	1–8	.125	5.	5.1	1–8	.12	4.80	4.90	11
12	7–64	.109375	4.375	4.4625	7–64	.109	4.36	4.45	12
13	3–32	.09375	3.75	3.825	3–32	.095	3.80	3.88	13
14	5–64	.078125	3.125	3.1875		.083	3.32	3.39	14
15	9–128	.0703125	2.8125	2.86875	5–64	.072	2.88	2.94	15
16	1–16	.0625	2.5	2.55	1–16	.065	2.60	2.65	16
17	9–160	.05625	2.25	2.295		.058	2.32	2.37	17
18	1–20	.05	2.	2.04		.049	1.96	2.00	18
					3–64				
19	7–160	.04375	1.75	1.785		.042	1.68	1.71	19
20	3–80	.0375	1.50	1.53		.035	1.40	1.43	20
21	11–320	.034375	1.375	1.4025	1–32	.032	1.28	1.31	21
22	1–32	.03125	1.25	1.275		.028	1.12	1.14	22
23	9–320	.028125	1.125	1.1475		.025	1.00	1.02	23
24	1–40	.025	1.	1.02		.022	.88	.90	24
25	7–320	.021875	.875	.8925		.02	.80	.82	25
26	3–160	.01875	.75	.765		.018	.72	.73	26
27	11–640	.0171875	.6875	.70125	1–64	.016	.64	.65	27
28	1–64	.015625	.625	.6375		.014	.56	.57	28
29	9–640	.0140625	.5625	.57375		.013	.52	.53	29
30	1–80	.0125	.5	.51		.012	.48	.49	30
31	7–640	.0109375	.4375	.44625		.01	.40	.41	31
32	13–1280	.01015625	.40625	.414375		.009	.36	.37	32
33	3–320	.009375	.375	.3825		.008	.32	.33	33
34	11–1280	.00859375	.34375	.350625		.007	.28	.29	34
35	5–640	.0078125	.3125	.31875		.005	.20	.20	35
36	9–1280	.00703125	.28125	.286875		.004	.16	.16	36
37	17–2560	.00664062	.265625	.2709375					37
38	1–160	.00625	.25	.255					38

Table 28-62. Measures of Weight, Contents and Area

Long Measure

12 inches = 1 foot.
3 feet = 1 yard.
5½ yards = 1 rod.
4 rods = 1 chain.
10 chains = 1 furlong.
8 furlongs = 1 mile.

Square Measure

144 square inches = 1 square foot.
9 square feet = 1 square yard.
30¼ square yards = 1 square rod.
160 square rods = 1 acre.
640 acres = 1 square mile.

Cubic Measure

1728 cubic inches = 1 cubic foot.
27 cubic feet = 1 cubic yard.
24.75 cubic feet = 1 perch.
128 cubic feet = 1 cord.

Liquid Measure

4 gills = 1 pint.
2 pints = 1 quart.
4 quarts = 1 gallon.
31½ gallons = 1 barrel.
2 barrels = 1 hogshead.

Avoirdupois Weight

16 ounces = 1 pound.
100 pounds = 1 hundredweight.
20 cwt. = 1 ton.

Table 28-63. Mensuration of Surfaces and Volumes

Area of rectangle = length × breadth.
Area of triangle = base × ½ perpendicular height.
Diameter of circle = radius × 2.
Circumference of circle = diameter × 3.1416.
Area of circle = square of diameter × .7854.

$$\text{Area of sector of circle} = \frac{\text{area of circle} \times \text{number of degrees in arc.}}{360}$$

Area of surface of cylinder = circumference × length + area of two ends.
To find the diameter of circle having given area: Divide the area by .7854, and extract the square root.
To find the volume of a cylinder: Multiply the area of the section in square inches by the length in inches = the volume in cubic inches. Cubic inches ÷ 1728 = volume in cubic feet.
Surface of a sphere = square of diameter × 3.1416.
Solidity of a sphere = cube of diameter × .5236.
Side of an inscribed cube = radius of a sphere × 1.1547.
Area of the base of a pyramid or cone, whether round square or triangular, multiplied by one-third of its height = the solidity.
Diam. × .8862 = side of an equal square.
Diam. × .7071 = side of an inscribed square.
Radius × 6.2832 = circumference.
Circumference = 3.5446 × $\sqrt{\text{area of circle.}}$
Diameter = 1.1283 × $\sqrt{\text{area of circle.}}$
Length of arc = no. of degrees × .017453 radius.
Degrees in arc whose length equals radius = 57° 2958′.
Length of an arc of 1 deg. = radius × .017543.
Length of an arc of 1 min. = radius × .0002909.
Length of an arc of 1 sec. = radius × .0000048.

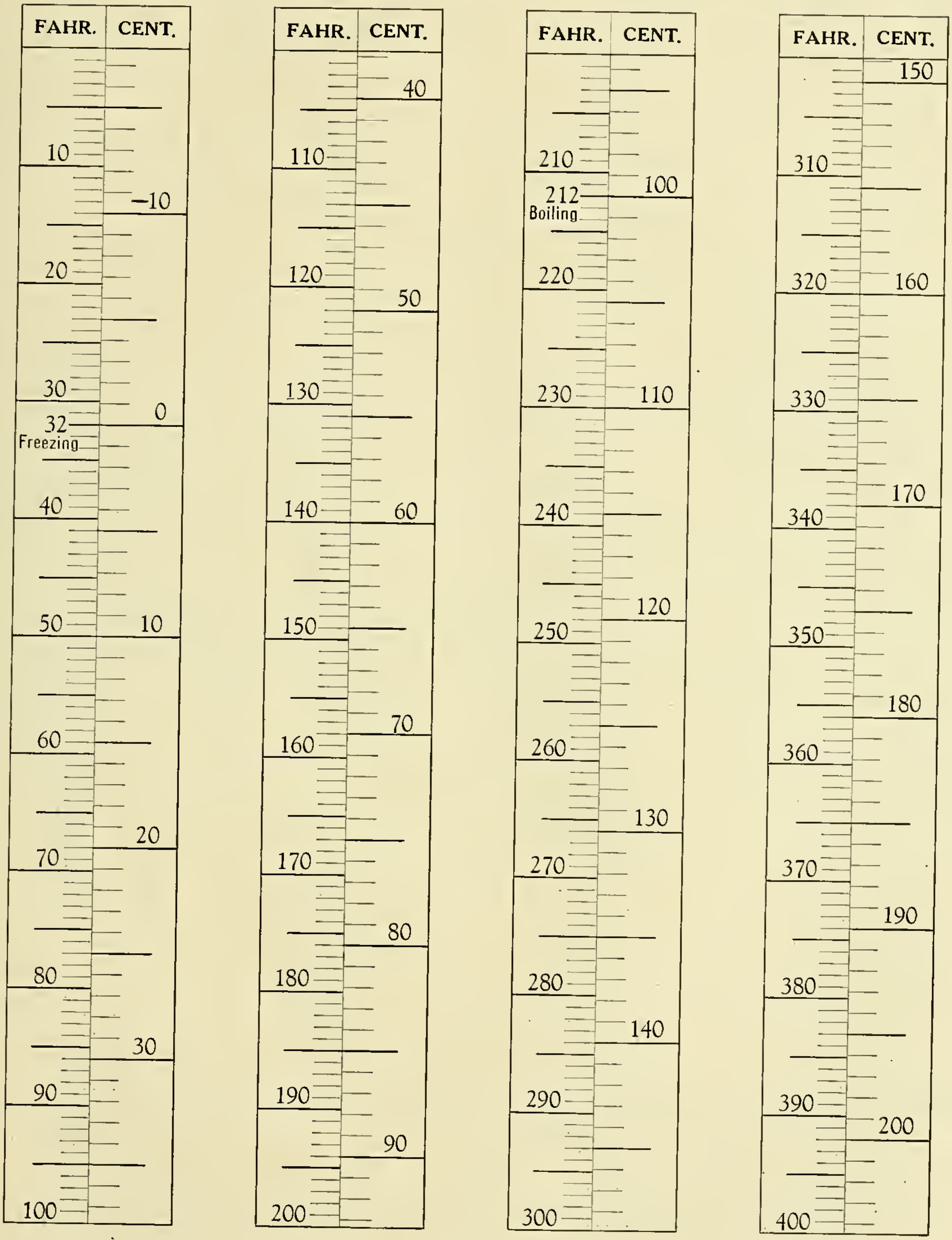
π = proportion of circumference to diameter = 3.1415926.
$\pi^2 = 9.8696044.$
$\sqrt{\pi} = 1.7724538.$
Log. $\pi = 0.49715.$
$1/\pi = 0.31831.$
$1/360 = .002778.$
$360/\pi = 114.59.$

Table 28-64. Electrical Units

Volt—The unit of electrical motive force. Force required to send one ampere of current through one ohm of resistance.
Ohm—Unit of resistance. The resistance offered to the passage of one ampere, when impelled by one volt.
Ampere—Unit of current. The current which one volt can send through a resistance of one ohm.
Coulomb—Unit of quantity. Quantity of current which, impelled by one volt, would pass through one ohm in one second.
Farad—Unit of capacity. A conductor or condenser which will hold one coulomb under the pressure of one volt.
Joule—Unit of work. The work done by one watt in one second.
Watt—The unit of electrical energy, and is the product of ampere and volt. That is, one ampere of current flowing under a pressure of one volt gives one watt of energy.
One electrical horsepower is equal to 746 watts.
One kilowatt is equal to 1,000 watts.
To find the watts consumed in a given electrical circuit, such as a pump motor, multiply the volts by the amperes.
To find the volts, divide the watts by the amperes.
To find the amperes, divide the watts by the volts.
To find the electrical horsepower required by a motor, divide the watts of the motor by 746.
To find the mechanical horsepower necessary to generate the required electrical horsepower, divide the latter by the efficiency of the generator.
To find the amperes of a given circuit, of which the volts and ohms resistance are known, divide the volts by the ohms.
To find the volts, when the amperes and ohms are known, multiply the amperes by the ohms.
To find the resistance in ohms, when the volts and amperes are known, divide the volts by the amperes.

Table 28-71. Conversion of Fahrenheit and Centigrade Temperatures

Formulae: fahr. = $\frac{9}{5}$ cent. + 32 deg. cent. = ^{5} (fahr. — 32 deg.)



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